Feasibility Study for Sea State 5 Skin-To-Skin Cargo Transfer Operations

Performed under ONR BAA 01-023
SKIN-TO-SKIN CONNECTED REPLENISHMENT

Submitted by:

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Code 2820 Expeditionary/Logistics Systems

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1. Executive Summary

This report details the results of the Feasibility/Concept Study for Sea State 5 Skin-to-Skin Cargo Transfer Operations, performed by the Naval Surface Warfare Center, Carderock Division, in response to the ONR Broad Agency Announcement 01-023, Skin-to-Skin Connected Replenishment. Skin-to-skin cargo transfer is a concept for transferring cargo within a "seabasing" environment. Seabase is a loosely defined term that refers to a collection of ships at sea conducting operations that enable forces to operate ashore without a large logistics footprint. Such an operation may require a wide variety of ships to transfer cargo such as pallets, containers, and vehicles to one another. The skin-to-skin concept would involve these ships pulling alongside and mooring together at very close proximity. The goal of this analysis was to assess the feasibility of skin-to-skin sea state 5 cargo transfer by focusing on three distinct subject areas; operational concepts, hydrodynamic modeling and simulation, and enabling technology identification and assessment.

As a result of the analysis, sea state 5 skin-to-skin connected replenishment is considered feasible, if the proper combinations of the concepts presented in this paper are implemented. These operations can only be performed with a "system of systems" approach. Mooring and fendering, cargo transfer, and ship control technologies must be selectively implemented to form a comprehensive system. In addition to the technology and concepts described in this analysis, full skin-to-skin capability would require a better definition of the operations and vessels to be involved.

2. Introduction

2.1. Background information

This report details the results of the Feasibility Study for Sea State 5 Skin-to-Skin Cargo Transfer, performed by the Naval Surface Warfare Center, Carderock Division, in response to the ONR Broad Agency Announcement 01-023, Skin-to-Skin Connected Replenishment. The study was performed with the requirements of the BAA as the primary guidance. The BAA is attached as Appendix A.

2.2. Analysis approach

The purpose of this study was to assess the feasibility of sea state 5 skin-to-skin replenishment cargo transfer operations. In addition, any enabling technology, concepts, and procedures are identified. The analysis for this project focused on three distinct subjects:

- Operational concepts
- Hydrodynamic modeling and simulation
- Enabling technology assessment

These three subjects are each important to understand for evaluating the feasibility of skin-to-skin operations.

The BAA called for a focus on the feasibility of skin-to-skin operations for ships in the seabase, including the MPF (F) (Maritime Preposition Force - Future), T-AKE and other Combat Logistics Force (CLF) ships, Amphibious Readiness Group (ARG), combatant vessels, commercial container and Roll On/Roll Off ships in conditions of at least sea state 3 (SS3) and up to sea state 5 (SS5). The BAA made no mention of the types of cargoes to be transferred, or the particular ships that are envisioned going skin-to-skin to transfer this cargo. In order to provide a framework for the recommendations in this report, a study of the previously mentioned ships, their likely cargo, cargo handling capabilities, and interface issues was performed. A logical set of ship combinations and cargo loads to be transferred has been developed, allowing the recommendations to be structured in a logical manner.

In order to make any recommendations about feasibility, the behavior of the above ships in the stated sea conditions must be understood. No body of information was available prior to this study that analyzed the specific situations that will be encountered in skin-to-skin operations, namely ships in a skin-to-skin configuration with forward speed. Various tools were investigated, with the Large Amplitude Motion Program (LAMP) being chosen as the most suitable for this study. SAIC was tasked to model a variety of ship-to-ship combinations under a selected sea conditions, forward speeds, and headings. In addition to the LAMP modeling, NSWCCD Seakeeping Department (Code 55) was tasked to use the WAMIT code to analyze a situation for comparison with the LAMP results. The details of the analyses and the results are presented in following sections of the report.

At the beginning of the project a number of capabilities possibly required for skin-to-skin operations was developed, and from this list potentially useful technologies were identified. Each technology has been evaluated to determine its applicability. This report will include descriptions of these technologies, possible applications of them, their capabilities and limitations, and a discussion of the current state of the technology. Some are immediately available as commercial equipment, some would require more developed engineering to be suitable for skin-to-skin operations, and others will require further basic research before a conclusive decision can be made.

After the operational concepts, hydrodynamics, and enabling technologies are discussed; example arrangements and combinations of systems and equipment that will enable skin-to-skin operations are presented. These examples illustrate the types of equipment, procedures, and planning that are required to conduct a ‘typical’ skin-to-skin operation.

3. Operational Concepts

As previously mentioned, the ONR BAA set no firm guidelines on what "skin-to-skin connected replenishment for ships in the seabase" meant. The only guidance provided was in the form of a list of ships to be considered, the transfer of dry and liquid cargo, and the specific mention of conducting operations between ships of widely varying sizes. The overall goal for conducting skin-to-skin operations is to enable seabasing operations. "Seabase" is a loosely defined term that refers to a collection of ships at sea conducting operations that enable forces to operate ashore without a large logistics footprint. Such an operation may require a wide variety of ships to transfer cargo such as pallets, containers, and vehicles to one another. The ships discussed in this section provide a representative sample of the types likely to be involved in a seabase.

A basic seabasing scenario has a group of ships acting as the seabase supporting operations ashore. This would likely consist of prepositioning ships including the MPF (F), amphibious ships, CLF, and combatants. These ships would require periodic resupply of cargo, both for their own consumption and for resupplying deployed forces. This cargo would be delivered by a variety of ships, including commercial containerships, other prepositioning ships, and ships of the Ready Reserve Force. These ships could either join the seabase and deliver the cargo to the appropriate ship(s), or they could deliver their cargo to one or more ships for storage and/or redistribution. It is the specifics of these interactions that must be developed in order to determine exactly what ship-to-ship combinations will be required, and therefore what technologies are necessary.

In order to put the research presented in this study in context, a set of general operational concepts and examples has been developed. This section presents ship combinations that could logically conduct skin-to-skin operations, the types of cargoes these ships would be capable of handling, and examples of possible interfaces to allow a conceptual understanding of how a skin-to-skin operation might look, and what technologies and procedures would be needed.

Tables 3.1 and 3.2 present the likely ship combinations and cargo types to be carried for skin-to-skin operations.

Table 3.1 – Ship-to-Ship Combinations

		RECEIVING							
		DDG/CG/DD(x)	LPD	LHD	CONTAINER	MPF(F)	T-AKR/LMSR	T-ACS	T-AKE
SUPPLY	DDG/CG/DD(x)	NO	NO	NO	NO	NO	NO	NO	NO
	LPD	NO	YES	YES	NO	NO	NO	NO	NO
	LHD	NO	YES	YES	NO	NO	NO	NO	NO
	CONTAINER	NO	YES	YES	NO	YES	YES	YES	NO
	MPF(F)	YES	YES	YES	YES	YES	YES	YES	YES
	T-AKR/LMSR	YES	YES	YES	YES	YES	YES	YES	YES
	T-ACS	YES	YES	YES	YES	YES	YES	YES	YES
	T-AKE	YES	YES	YES	NO	YES	YES	YES	YES

Table 3.2 – Cargo Compatibility

		TYPE OF CARGO THAT CAN BE HANDLED ONBOARD			
		PALLET	QUADCON	TEU	VEHICLES
SHIP TYPE	DDG/CG/DD(x)	YES	NO	NO	NO
	LPD	YES	YES	YES	YES
	LHD	YES	YES	YES	NO
	CONTAINER	NO	NO	YES	YES
	MPFF	YES	YES	YES	YES
	T-AKR/LMSR	YES	YES	YES	YES
	T-ACS	YES	YES	YES	YES
	T-AKE	YES	YES	NO	NO

These tables show how ships could function as consumers, cargo deliverers, and in some cases both. For instance, combatants such as the DDG-51 or CG-47 have no capacity to handle cargo larger than pallet size, and lack the equipment to transfer cargo to another ship. Therefore, their role in any skin-to-skin operation would be as a receiver of cargo. On the other end of the spectrum is the MPF (F), which is capable of handling the full range of possible cargo, and having a capacity to deliver this cargo to any ship in the seabase.

The unique capability provided by skin-to-skin operations is that a great variety of cargo can be handled. UNREP using the STREAM system is currently limited to approximately 5,700 lbs (~2.5 LT). R&D is ongoing to increase that capability to 12,000 lbs (~5.4 LT). By conducting the operation skin-to-skin, there is the possibility for using cargo handling equipment with much greater capacity than the STREAM system. Shipping containers, wheeled and tracked vehicles, and outsized loads can be easily handled by a variety of shipboard cranes. Additionally, the small distances involved in skin-to-skin operations would enable current UNREP equipment to transfer palletized cargo and quadcon sized containers more quickly.

The following examples in Figures 3.1 - 3.5 are meant to be both specific and general. For instance, Figure 3.1 shows an LMSR and T-ACS ship. These are both currently deployed ships that could conceivably be involved in a skin-to-skin operation. However, the LMSR also represents a possible configuration and size of what the MPF (F) could be like, and is also similar in size to a large containership. The T-ACS also represents a small to medium sized containership.

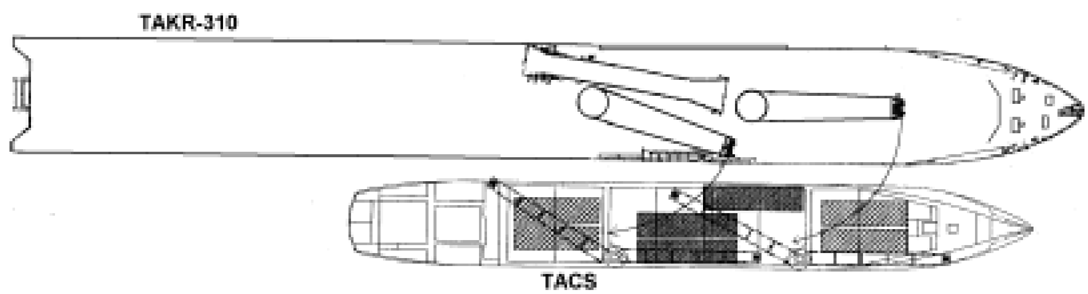


Figure 3.1 – LMSR & T-ACS

The LMSR and T-ACS are each capable of handling the full range of cargo likely to be involved in a skin-to-skin operation. The LMSR is a RO/RO vessel capable of handling all types of cargo, the T-ACS is a converted container/breakbulk ship that can carry vehicles, either in its holds on flatracks or lashed to the deck. Both are equipped with cranes suitable for delivering cargo to a ship moored alongside. Possible scenarios include an LMSR delivering containers and vehicles on flatracks to a T-ACS ship for transfer to a JLOTS operation, or one of the ships acting as a shuttle ship, delivering new supplies to the seabase. An alternative scenario has the LMSR representing the MPF (F) taking on a supply of containers from a commercial containership.

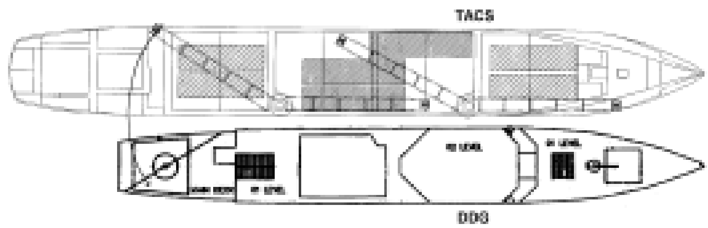


Figure 3.2 – T-ACS & DDG-51

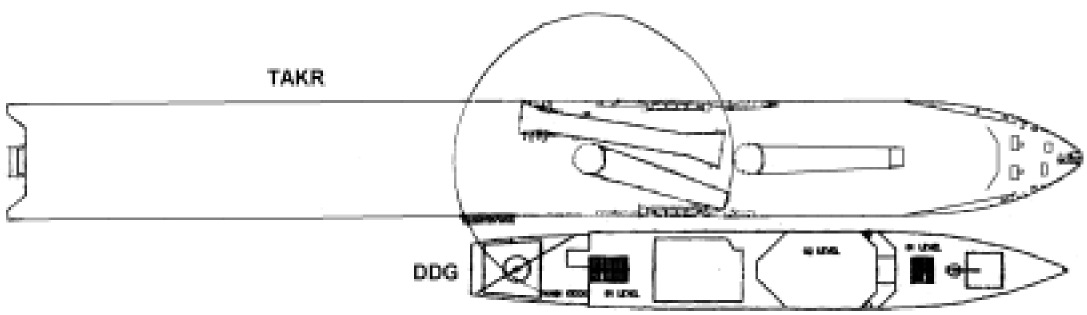


Figure 3.3 – LMSR & DDG-51

Figures 3.2 and 3.3 show how cargo ships such as the LMSR and T-ACS could be used in a resupply role for combatants. In most cases UNREP is entirely sufficient for transferring the small loads required by combatant craft. Only in the case of outsized cargo is UNREP not suitable. However, by utilizing large capacity cranes and open deck spaces, such as the vertical replenishment (VERTREP) areas on a DDG, large numbers of small pallet loads could be transferred very quickly. A single LMSR crane has a capacity of 57 LTs, which could enable it to deliver a high volume of cargo very quickly. The T-ACS cranes have a capacity of 30 LTs. Another unique capability could be to remove damaged or otherwise incapacitated helicopters from the ship, making room for a replacement. Finally, at-sea rearming of the combatant's Vertical Launch System (VLS) tubes could be possible using the LMSR or T-ACS cranes. It is situations and unique capabilities such as these that must be well defined in order to determine exactly what types of systems will be needed.

For example, the T-ADC(X) (now T-AKE) performance specification lists example UNREP scenarios. In one, a general stores resupply of a DDG, a 56 pallet load is allotted 2 hours for transfer, not including the connect and disconnect times. Using the conservative assumption that these are 5,700lb full load pallets, a T-ACS-style crane could deliver the entire load with approximately 5 crane lifts, assuming the existence of a device, such as a flatrack, that would allow the crane to lift many pallets simultaneously. In reality, the pallets would have a lower average weight than 5,700 lbs, requiring fewer crane lifts. The potential for transferring a large amount of cargo in a short time is evident.

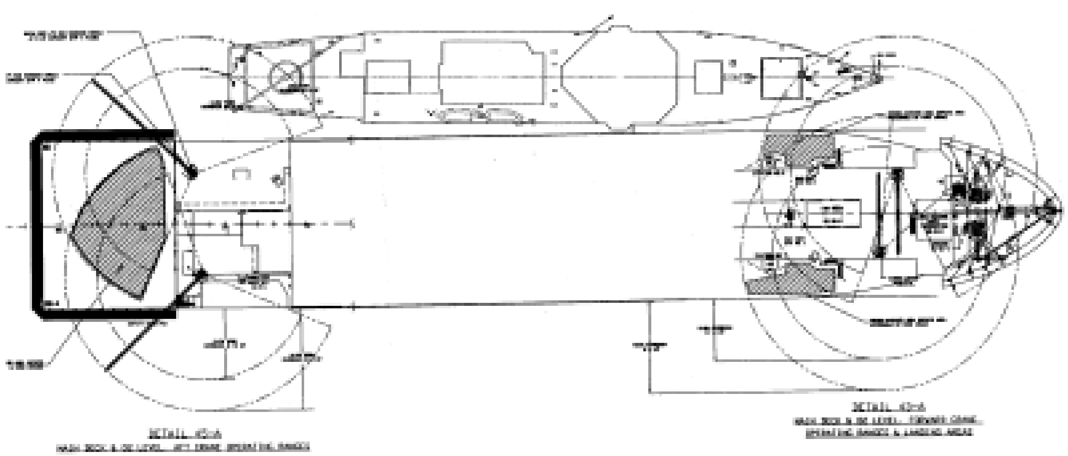


Figure 3.4 – T-AKE & DDG-51

Figure 3.4 shows a T-AKE and DDG-51, showing how a conventional UNREP ship could transfer it's cargo skin to skin instead of at the normal separation distance approximately 150' with it's UNREP gear. Using the STREAM system at these short distances would allow for very quick transfer of the usual palletized loads. The new T-AKE UNREP ship is also equipped with cargo cranes forward and aft, on both sides of the ship. These cranes are designed for loading palletized cargo from a pier and are considerably smaller than those found on the T-ACS or LMSR. They have a 9.8 LT capacity at 12 m outreach, and 4.9 LT at 18m outreach. This could provide a supplemental capability to the standard UNREP system.

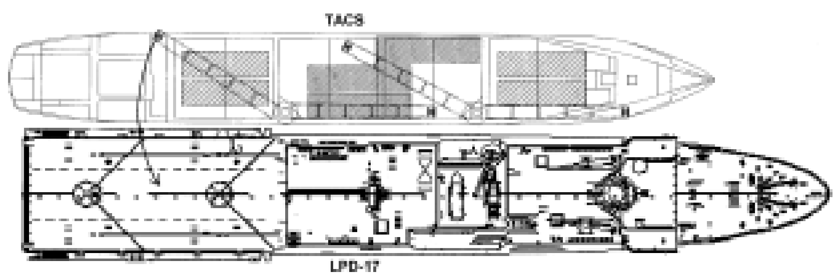


Figure 3.5 – T-ACS & LPD-17

Figure 3.5 shows the interface between a T-ACS and LPD-17. The LPD-17 has a large flight deck that would be ideally suited for receiving large cargo items, or large numbers of smaller items such as pallets. The LPD-17 is a ship that would be involved in amphibious operations, and could potentially receive vehicles and other equipment that would not normally be involved in UNREP. The ability to resupply its mission cargo in theater would be a significant benefit of skin-to-skin operations.

The examples above are only a fraction of the potential applications of skin-to-skin cargo transfer. They highlight several of the unique challenges to conducting these operations as well. In the cases involving the LMSR, or future ships like it, a significant interface issue exists due to that ship's 50' freeboard. In the case of the DDG-51, the hydrodynamic studies show significantly higher roll motions than for the other ships. This presents a unique problem for operations including this type of ship that may not be present when other ships are involved. It also highlights the need to develop more detailed operational concepts before doing any system design work. The inclusion or exemption of certain ships or unique capabilities could have a drastic impact on the requirements of the systems involved in the operation.

4. Hydrodynamic Analysis

One of the objectives of this feasibility analysis was to quantify the ship motions expected in the sea state 5 skin-to-skin replenishment scenario. Two hydrodynamic analyses were conducted in order to investigate the motion and behavior of selected sealift and combatant ships operating in a skin-to-skin scenario. The results were used to determine that the ship motion was not a limiting factor in skin-to-skin operations.

The hydrodynamic analysis consisted of two separate but related efforts. The primary focus was on a regular and irregular waves study conducted using LAMP software. A similar WAMIT study was also conducted to validate the LAMP results.

4.1. LAMP Analysis

4.1.1. The LAMP Method

The primary ship motion and force analysis for this study were solved numerically using the three-dimensional time-domain Large Amplitude Motions Program (LAMP) developed by SAIC. LAMP is a multi-level system that deals with different levels of nonlinearity in seakeeping problems. In this evaluation the LAMP Multi-Body code (LAMP-MB 2.9.1), using the LAMP-2 method was used.

In the LAMP-2 method, the nonlinear hydrostatic restoring forces and the Froude-Krylov forces are computed accurately by integrating the pressure distribution on the wetted area of the hull surface below the incident waves. The hydrodynamic forces are computed on ship's linear position in the seaway. In the multi-body version of LAMP, the dynamics of each body is computed and tracked separately, but the hydrodynamics solution is unified. The influence of the diffraction and radiation by each body on the other is included in the solution.

All the computed external forces and moments are applied to the center of gravity of the ship, and then the equation of motions in six degrees of freedom is solved. In this analysis the forces for the constrained motions are output, and the motions in the unconstrained modes are output.

4.1.2. Constraints and Controls

Any calculation in other than long crested head seas introduces transverse (in-plane) forces and moments on the ship. As in a real seaway, the ship must employ control forces to maintain heading and course. In the LAMP simulation any control force such as a rudder can be modeled as an external force in the dynamic simulation. In order to allow a stand-alone simulation, an automatic control system that measures heading and course and applies rudder commands must also be implemented.

While such control models can be implemented in LAMP, the subject ships did not define a control algorithm that could maintain the ships adequately on course without collisions. An alternative is to fix the ship on course by constraining the yaw motion and the sway motion. This can be done by adding “soft” springs that limit the motion but allow some sway and yaw. In the limiting case, the sway and yaw can be completely restrained. An assumption in this analysis is that the ships will be constrained by control surfaces and ship-ship connections, and that the intent is to measure the magnitude of the forces these connections must support. Therefore, in all of the cases in this study, the ships were restrained in surge, sway and yaw, and free in roll, pitch and heave. The surge, sway and yaw restraining forces and moments were measured and represent the force and moment that must be imposed to keep the two ships separated.

4.1.3. Irregular Wave Analysis

All of the irregular wave analysis was carried out in random long crested seas, created by the superposition of a set of sinusoidal waves. The waves were generated to match the Bretschneider energy spectrum defined by the specified significant wave height and modal period. Frequencies for the regular waves were determined by a geometric spreading of the frequency intervals on either side of the modal period. Phases for the waves were random with a uniform distribution. The time domain analysis was carried out for 30 minutes of full-scale time.

For each of the physical properties measured, the following statistical analysis was performed. The mean of the process was calculated and subtracted from each step. All the zero crossings were tabulated, and the maxima and minima between zero-crossings found. The maxima and minima were sorted separately and together as absolute values, and the highest 1/3rd of the peaks were averaged, and the highest 1/10th of the peaks were averaged. These values were tabulated in the database, and the 1/3rd highest are presented graphically in this report for selected quantities.

4.1.4. Regular Wave Analysis

A limited set of skin-to-skin analysis was carried out in regular waves. For these cases sufficient encounter cycles were calculated to give a steady response for about 40 cycles. The initial non-regular part of the record was truncated, and the mean of the remaining peaks was averaged to represent the linearized response at that frequency.

4.1.5. Geometry and Particulars

4.1.5.1. Ship Descriptions

Four ships were modeled, including two large cargo ships, a smaller crane/container ship, and a naval combatant. These ships were chosen because they provide a good representation of the size and behavior of vessels that will potentially be involved in skin-to-skin operations. These ships were:

LMSR – a large Roll-On/Roll-Off ship with a non-immersed transom stern and a bow bulb.
MPF – a nominal Maritime Prepositioning Ship. This geometry is a geosim of the LMSR with length reduced and draft increased.
DDG – The DDG-51 Arleigh Burke Class Guided Missile Destroyer, a slender transom-stern naval combatant with a below-baseline sonar dome.
TACS – The T-ACS 5 Auxiliary Crane Ship, a moderately full form ship with a transom stern.

Figures 4.1 - 4.4 show a profile view a body plan view of each ship. The figures show the LAMP panelizations for the ships.

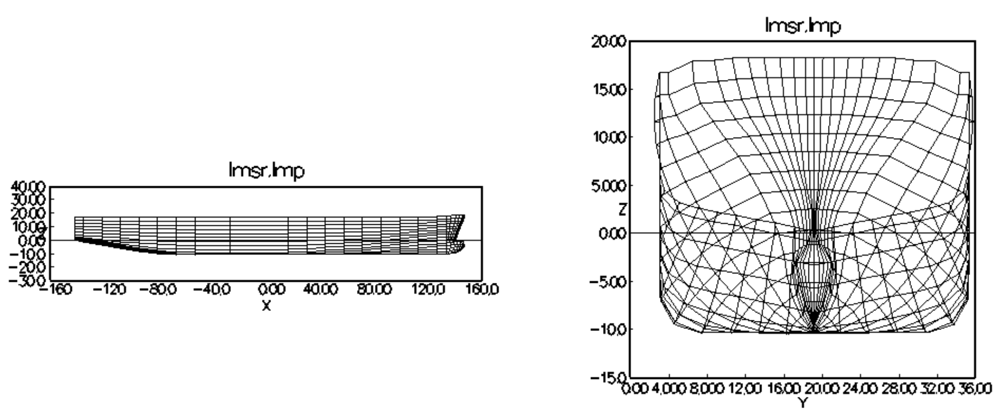


Figure 4.1 - LMSR Profile and Body Plan

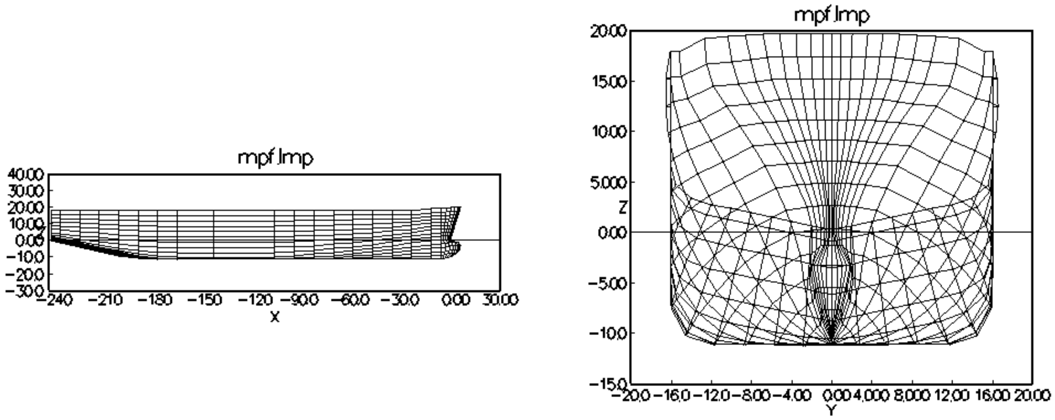


Figure 4.2 - MPF Profile and Body Plan

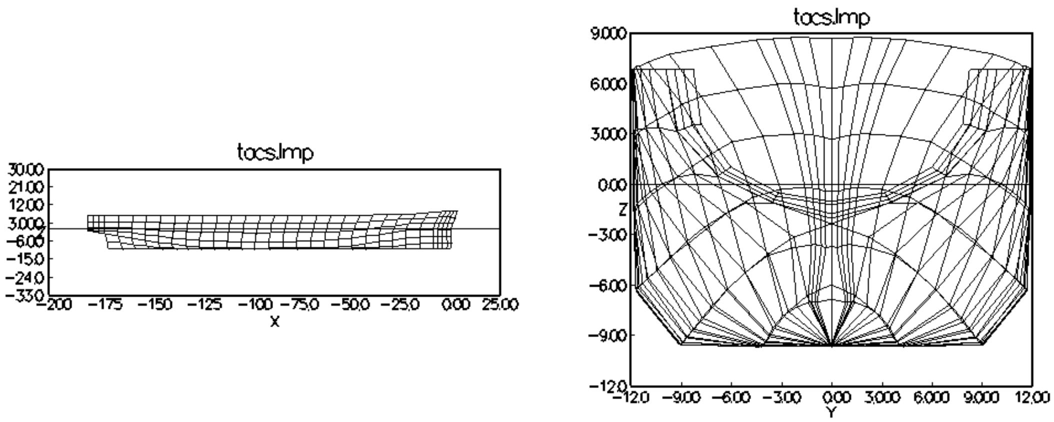


Figure 4.3 - TACS Profile and Body Plan

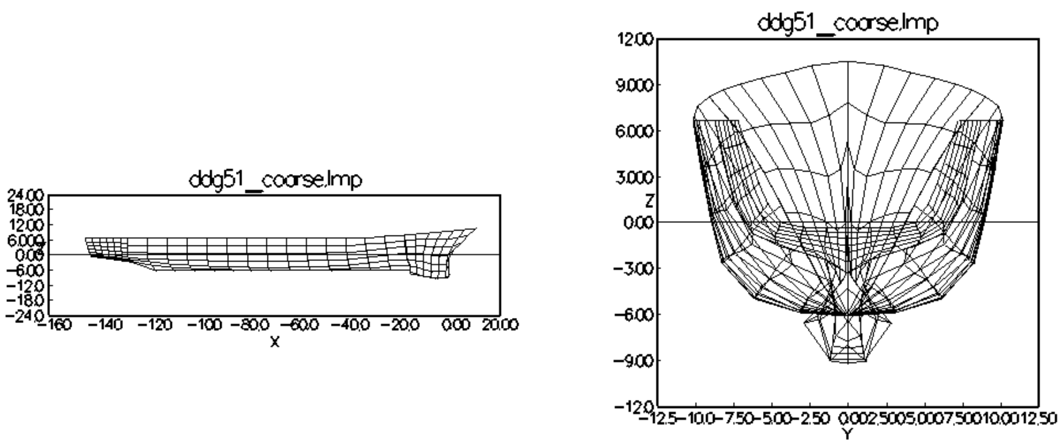


Figure 4.4 - DDG Profile and Body Plan

Table 4.1 shows geometric particulars for each ship.

Table 4.1 - Geometry Particulars for 4 Subject Ships

All Values in Meters	LMSR	MPF	TACS	DDG51
Wetted Surface Area S	11027.	9604.7	5610.7	2843.5
Submerged Volume V	61307.	55575.	25394.	7698.2
Longitudinal Center of Buoyancy LCB	-4.1308	-6.9934	-2.6012	-4.9838
Transverse Center of Buoyancy TCB	19.164	19.164	-15.016	-12.000
Vertical Center of Buoyancy VCB	-4.5746	-4.9110	-4.3768	-2.4826
Waterplane Area AWP	7607.5	6423.9	3329.7	1974.7
Longitudinal Center of Flotation LCF	-11.335	-13.092	-8.1690	-12.224
Longitudinal Metacentric Height BML	611.21	407.90	244.82	313.58
Transverse Center of Flotation TCF	19.164	19.164	-15.016	-12.000
Transverse Metacentric Height BMT	9.5313	8.8368	4.9669	5.2225
Vertical Center of Gravity (input) VCG	3.5000	1.5000	-0.67640	0.95390
Longitudinal Metacentric Height GML	603.14	401.49	241.12	310.15
Transverse Metacentric Height GMT	1.4567	2.4258	1.2665	1.7860
Waterline Length LWL	281.89	238.60	181.13	143.61
Waterline Maximum Beam B	32.233	32.157	23.714	17.841
Maximum Draft T	10.458	11.227	9.6260	9.1433

4.1.5.2. Ship – Ship Placement

In the description of the center of gravity position and the location of points for relative motions output and moments, the following ship-fixed coordinate system is used:

- X – positive forward, X = 0 at station 10, the center of the waterplane.
- Y – positive to port, Y=0 at the centerplane of the ship
- Z – positive upward, Z=0 at the mean waterline

In each case, the ships were separated by 20 feet, or 6.096 meters, skin-to-skin. The ships were aligned longitudinally with their Centers of Gravity aligned for most cases, and with Station 15 aligned for some cases. In each case the "second" ship, or smaller ship was placed to the starboard side of the first ship, and waves were defined as approaching the port side. This meant that the larger ship sheltered the smaller ship by diffracting some of the ambient waves. The second ship was defined in the coordinate system of the first ship.

Figures 4.5 – 4.12 show plan views and end views of the relative position for the CG aligned cases.

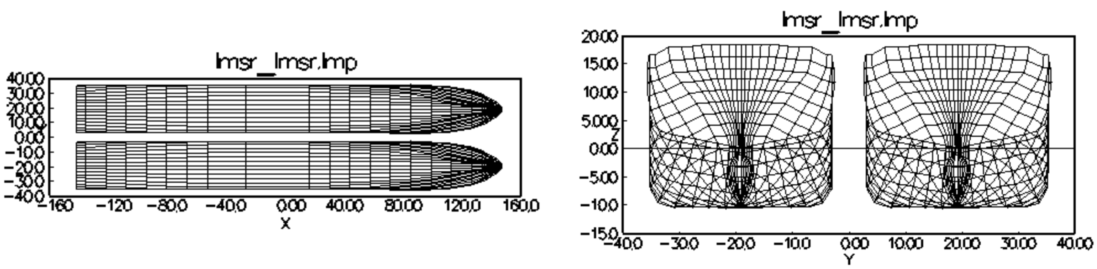


Figure 4.5 - LMSR as Ship #1 alongside LMSR as Ship #2, CG's Aligned

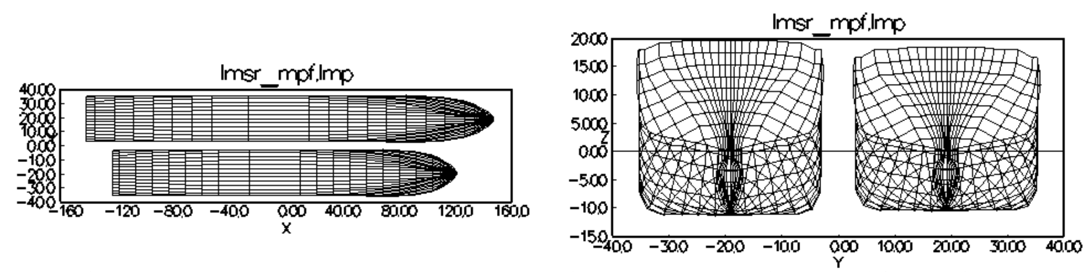


Figure 4.6 - LMSR as Ship #1 alongside MPF as Ship #2, CG's Aligned

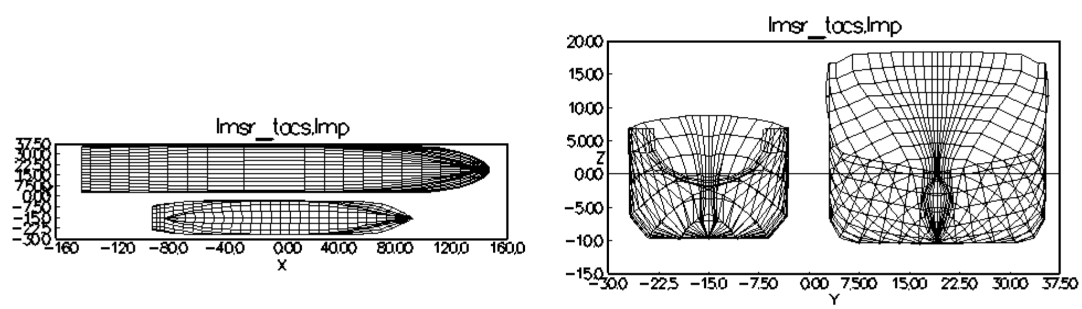


Figure 4.7 - LMSR as Ship #1 alongside TACS as Ship #2, CG's Aligned

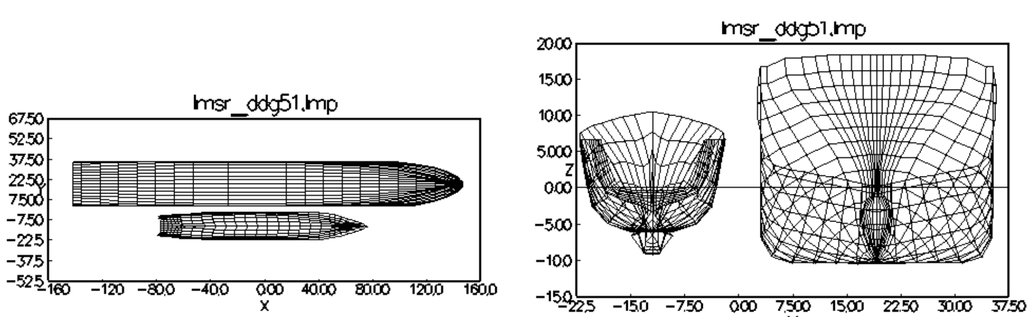


Figure 4.8 - LMSR as Ship #1 alongside DDG as Ship #2, CG's Aligned

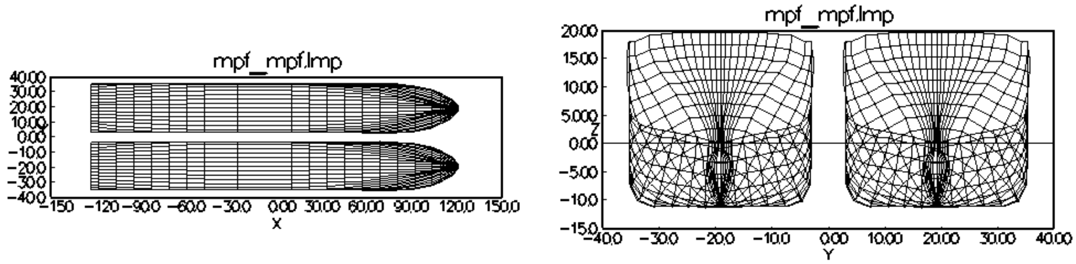


Figure 4.9 - MPF as Ship #1 alongside MPF as Ship #2, CG's Aligned

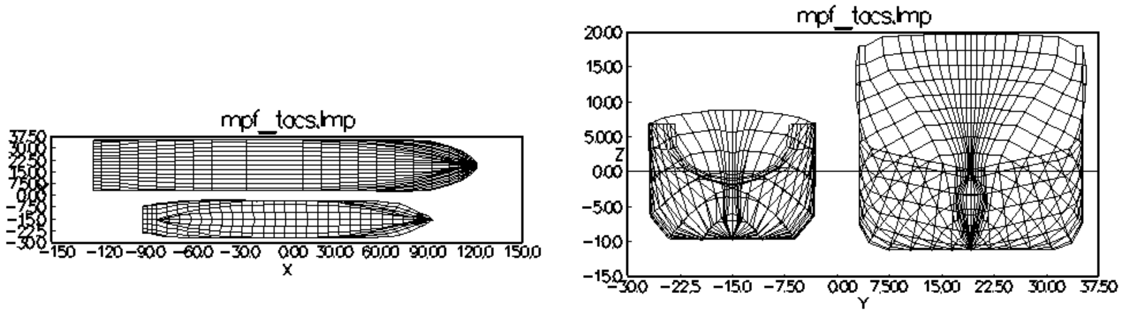


Figure 4.10 - MPF as Ship #1 alongside TACS as Ship #2, CG's Aligned

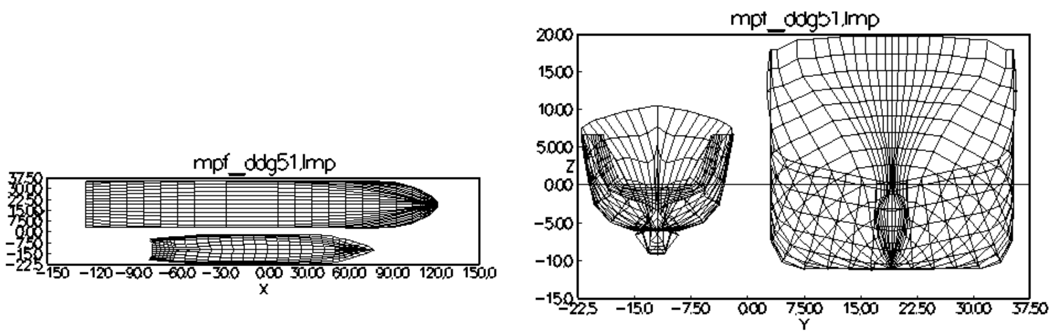


Figure 4.11 - MPF as Ship #1 alongside DDG as Ship #2, CG's Aligned

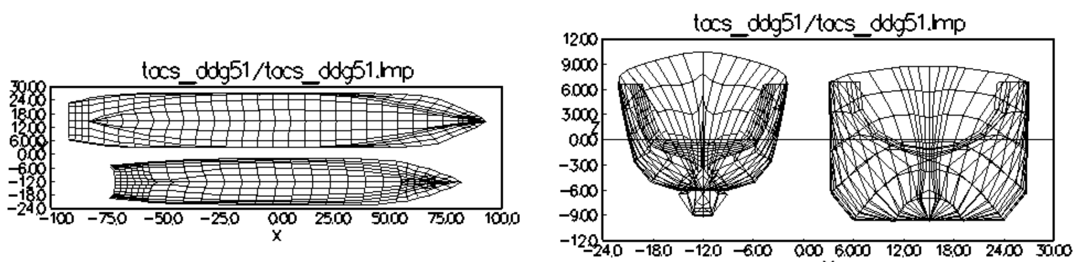


Figure 4.12 - TACS as Ship #1 alongside DDG as Ship #2, CG's Aligned

Figures 4.13 – 4.15 show the same views for all of the Station 15 aligned cases.

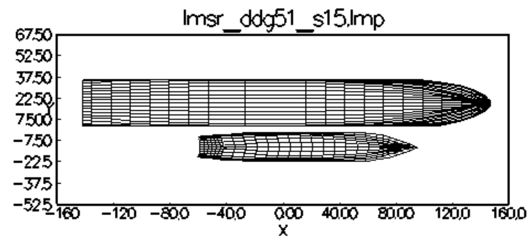


Figure 4.13 - LMSR as Ship #1 alongside DDG as Ship #2, Station 15 Aligned

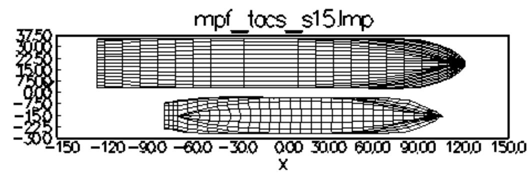


Figure 4.14 - MPF as Ship #1 alongside TACS as Ship #2, Station 15 Aligned

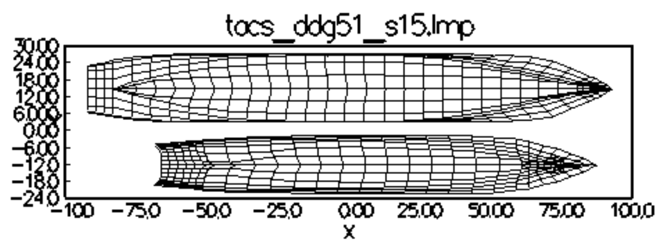


Figure 4.15 - TACS as Ship #1 alongside DDG as Ship #2, Station 15 Aligned

4.1.5.3. Relative Motion Points

For the relative motion calculation it was desired to choose 4 points along the deck edge that would correspond to a point on the deck edge of the other ship. Thus each ship-ship combination has separate set, shown in Tables 4.2 and 4.3.

Table 4.2 - Relative Motion Points for CG Aligned Cases

SHIP 1 (to Port, Sheltering)			*SHIP 2 (to Starboard, Sheltered)*		
LMSR			**LMSR**		
X, m	Y, m	Z, m	X, m	Y, m	Z, m
100	3	16.75	100	-3	16.75
40	3	16.75	40	-3	16.75
-60	3	16.75	-60	-3	16.75
-120	3	16.75	-120	-3	16.75
LMSR			**DDG**		
50	3	16.75	50	-3	8.2
25	3	16.75	25	-1.9	6.65
-25	3	16.75	-25	-1.9	6.65
-50	3	16.75	-50	-2.4	6.65
LMSR			**MPF**		
90	3	16.75	90	-4	18
50	3	16.75	50	-3	18
-50	3	16.75	-50	-3	18
-90	3	16.75	-90	-3	18
LMSR			**TACS**		
60	3	16.75	60	-3.75	7.0
40	3	16.75	40	-3.2	6.9
-40	3	16.75	-40	-3	6.8
-60	3	16.75	-60	-3.15	6.8
MPF			**MPF**		
90	4	18	90	-4	18
50	3	18	50	-3	18
-50	3	18	-50	-3	18
-90	3	18	-90	-3	18
MPF			**MPF**		
60	3	18	60	-3.75	7.0
40	3	18	40	-3.2	6.9
-40	3	18	-40	-3.0	6.8
-60	3	18	-60	-3.15	f 6.8
MPF			**DDG**		
50	3	18	50	-3	8.2
25	3	18	25	-1.9	6.65
-25	3	18	-25	-1.9	6.65
-50	3	18	-50	-2.4	6.65
TACS			**DDG**		
50	3.3	7.0	50	-3	8.2
25	3.0	6.8	25	-1.9	6.65
-25	3.0	6.8	-25	-1.9	6.65
-50	3.1	6.8	-50	-2.4	6.65

Table 4.3 - Relative Motion Points for Station 15 Aligned Cases

SHIP 1 (to Port, Sheltering)			*SHIP 2 (to Starboard, Sheltered)*		
TACS			**DDG**		
X, m	Y, m	Z, m	X, m	Y, m	Z, m
60	3.75	7.0	60	-3.2	8.4
40	3.2	6.9	40	-1.9	6.825
-25	3.2	6.8	-25	-2.0	6.65
-50	3.1	6.8	-50	-3.1	6.65
LMSR			**DDG**		
70	3	16.75	70	-3.4	8.6
40	3	16.75	40	-1.9	6.65
-10	3	16.75	-10	-1.9	6.65
-40	3	16.75	-40	-2.9	6.65
MPF			**DDG**		
75	3	18	75	-3.84	7.2
50	3	18	50	-3.2	6.9
-25	3	18	-25	-3.2	6.8
-50	3	18	-50	-3.25	6.8

4.1.5.4. Sea State Definitions

The US Navy and NATO define standard Sea States by a range of modal periods and wave heights. The following sea state definitions correspond to the higher end of the Sea State in the Navy/NATO standard definition. In this simulation the two parameters were used to define a Bretschneider spectrum. This spectrum was represented by 20 regular waves, which were superimposed to create the wave environment in the LAMP-MB simulation. Table 4.4 gives the sea state particulars used in this study.

Table – 4.4 Sea State Definitions

Sea State	Significant Wave Height, meters	Modal Period, Seconds
3	1.25	7.5
4	2.50	8.6
5	4.00	9.7

4.1.5.5. Run Conditions

Five sets of calculations were performed, each with different run conditions and outputs. All cases were run with a standoff distance of 20 ft.

1. Irregular Seas with CG Aligned (144 cases)
   - LMSR/LMSR, LMSR/MPF , LMSR/TACS , LMSR/DDG, MPF/MPF, MPF/DDG, MPF/TACS, TACS/DDG ship combinations
   - Sea States 3, 4, 5 (high end values in NATO Standard table)
   - Three wave headings (head, quartering, following)
   - Two speeds (8 and 16 knots)
   - CG's aligned

2. Irregular Seas with Station 15 Aligned (36 cases)
   - LMSR/DDG, TACS/MPF, DDG/TACS ship combinations
   - Sea States 4 and 5 (high end values in NATO Standard table)
   - Three wave headings (head, quartering, following)
   - Two speeds (8 and 16 knots)
   - Station 15 aligned

3. Validation Cases - Irregular Seas, Zero Speed (9 cases)
   - TACS/LMSR ship combination
   - Sea States 3, 4, 5 (high end values in NATO Standard table)
   - Three wave headings (head, quartering, following)
   - CG's aligned

4. Validation Cases - Regular Waves, Zero Speed (60 cases)
   - TACS/LMSR ship combination
   - Sea States 3, 5 (high end values in NATO Standard table)
   - Three wave headings (head, quartering, following)
   - CG's aligned
   - Twenty Frequencies with amplitude derived from Sea State spectra as shown in Table 4.5

Table 4.5 - Regular Waves Derived from Sea States for Validation Cases

Sea State 3		Sea State 5	
Wave Amplitude, meters	**Wave Period, Seconds**	**Wave Amplitude, meters**	**Wave Period, Seconds**
0.0025	2.95	0.0079	2.28
0.0319	3.48	0.1022	2.69
0.0717	3.91	0.2295	3.02
0.0933	4.25	0.2987	3.29
0.0988	4.53	0.3160	3.50
0.0956	4.75	0.3058	3.67
0.0887	4.92	0.2838	3.81
0.0807	5.06	0.2581	3.91
0.0726	5.17	0.2324	4.00
0.0651	5.26	0.2082	4.07
0.0840	5.37	0.2689	4.15
0.1076	5.55	0.3442	4.29
0.1344	5.85	0.4300	4.52
0.1588	6.35	0.5082	4.91
0.1691	7.19	0.5412	5.56
0.1534	8.60	0.4910	6.65
0.1144	10.96	0.3661	8.47
0.0702	14.91	0.2247	11.52
0.0366	21.52	0.1170	16.64
0.0168	32.60	0.0537	25.21

5. Regular Wave Cases (36 cases)
   - TACS/LMSR ship combination
   - Wave Periods of 12, 14, 16 and 18 seconds
   - Wave Amplitudes of 1.0, 1.5 and 2.0 meters
   - Three wave headings (head, quartering, following)
   - CG's aligned

4.1.6. Results

The results of the motion analysis varied depending on the ship configurations, wave conditions, and heading. Due to the large amount of data generated during this analysis only selected summary plots are listed in this section. The cases that are most relevant to the SS5 skin-to-skin scenario (SS5, CG's aligned, 8 and 16 knots, 3 headings) are included in the attached appendix. If desired, similar plots can be generated for all test conditions.

The regular wave cases in which the wave height was varied generally showed a trend towards linear behavior. There are several exceptions, in particular the roll in following, regular seas for the LMSR next to the TACS show a very significant non-linearity for the LMSR. The TACS also displays some nonlinearity in roll. As might be expected, this case generated some of the largest relative motion in any of the cases we studied showing a relative motion of almost 3.5 meters in a 2 meter wave. In the rest of the study, the relative motion was less than one meter.

The motion and force data indicate that, while sealift and combatant ships operating in a skin-to-skin configuration do experience significant motion in a SS5, the motion and the resulting force are not the limiting factor proposed skin-to-skin operations. The relative motion in the ships, along with deficiencies in current operating procedures and hardware, can be neutralized with the technology and procedures detailed in later sections of this feasibility study.

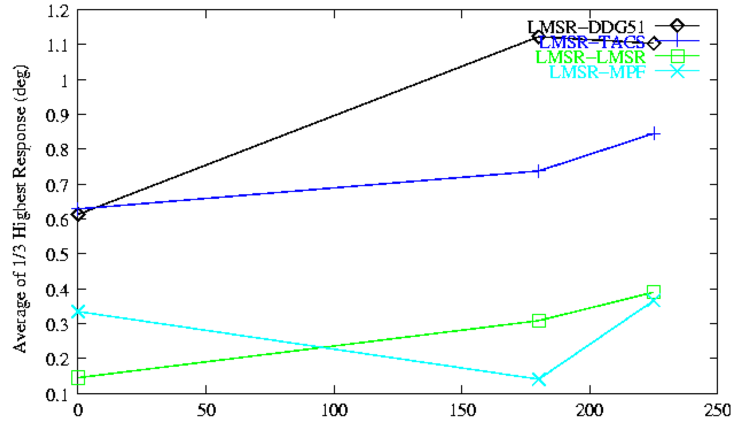


Figure 4.16 – Ship-to-Ship Effect on Relative Motion
LMSR w/4 ships, SS5, 8 knots

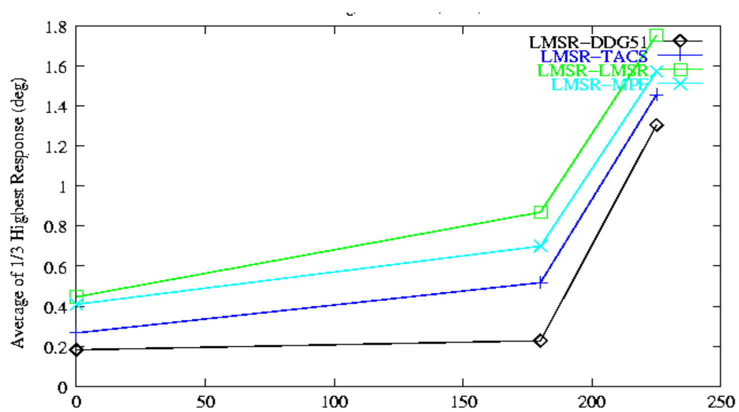


Figure 4.17 – Ship-to-Ship Effect on Roll
LMSR w/4 ships, SS5, 8 knots

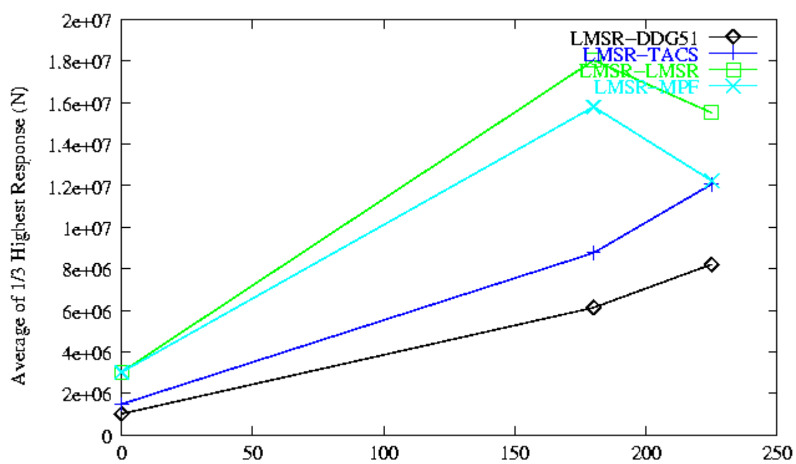


Figure 4.18 – Ship-to-Ship Effect on Sway Force
LMSR w/four ships, SS5, 8 knots

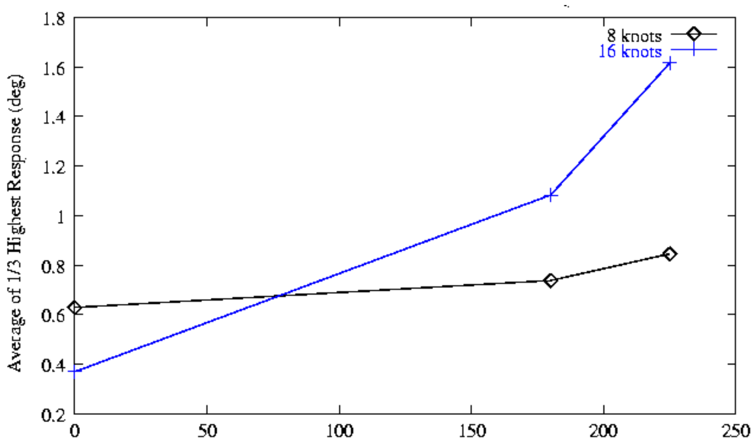


Figure 4.19 – Speed Effect on Relative Motion
T-ACS Roll w/ LMSR in SS5

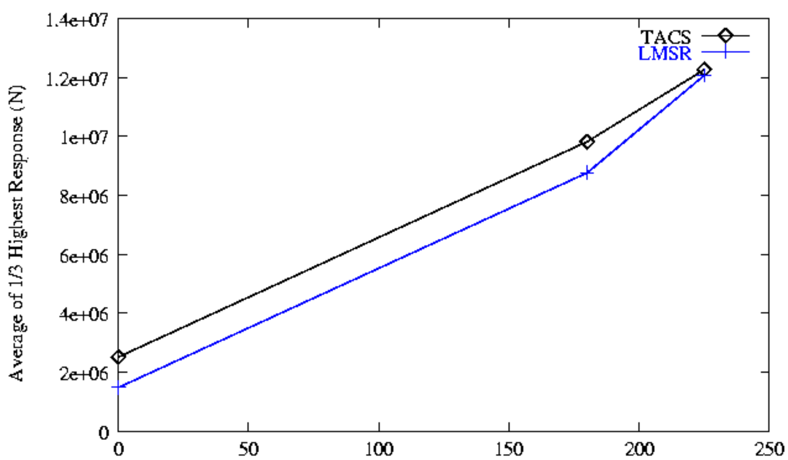


Figure 4.20 – Mutual Sway Force
LMSR and T-ACS Induce Similar Force on Each Other, SS5, 8 knots

4.2. WAMIT Analysis

Because the LAMP software used in the primary motion analysis has yet to be validated with empirical data in a multi-body, at speed scenario, similar calculations were performed for comparison using WAMIT, software that has been previously validated. Because WAMIT is accurate in a zero speed conditions only, both LAMP and WAMIT conducted zero speed runs using the same environmental conditions listed above. The T-ACS ship and the LMSR are the vessels used in the validation cases.

4.2.1. WAMIT Model

As a first step, Ship Motion Computer Program (SMP) input files were generated for each ship. The T-ACS file has a skeg, bilgekeel, and rudder for appendages; the LMSR file has a skeg and bilgekeel. The SMP file format is convenient format for further manipulation and provides a check on WAMIT hydrostatics and viscous roll damping.

The SMP input files were converted to MultiSurf files for grid generation. The bow bulb, transom (where needed), and skeg were added as separate surfaces. A working MultiSurf file allows for quick re-gridding for convergence studies and multiple export formats.

A WAMIT Geometry Definition File (GDF) was generated from the MultiSurf grids. The T-ACS 5 was modeled with 962 panels/side; the LMSR with 1,820 panels/side. Each ship has one plane of symmetry, i.e., the centerline. The panels are typically 10 ft long with 18-29 panels from keel to waterline. Care was taken to make most of the panels roughly rectangular. A convergence study was done to determine an acceptable number of panels.

WAMIT hydrostatics for both ships were checked with SMP hydrostatics and found correct. Extra linear roll damping was added to both ships based on values from the SMP output appropriate for the roll levels predicted by WAMIT. Considering the separation between wave excitation and natural roll period, the increased damping has little effect.

Five wave headings were examined – 180 (head) to 0 (following) in 45 degree increments. Due to the WAMIT wave heading convention and coordinate system, the LMSR was placed to the starboard of the TACS. In this position, the LMSR shields the TACS from incoming waves. The centerline-to-centerline distance is 112 ft. The runs were made for infinite water depth.

The ships were connected by a 6 degree-of-freedom spring located at the LCB/LCG, the ships' half beam, and 20.42 ft above the free surface. The half beam locations are –39 ft for the TACS and 53 ft for the LMSR. The connections are considered infinitely rigid. Using the half beam of the ships for a connection location is equivalent to using the midpoint between the centerlines for sway and yaw restraint. They are nearly the same for surge, sway, and yaw restraint connectors, but the half beam position results in less relative surge.

Use of the 6DOF spring allows for total restraining force and moment to be easily calculated. Previous Code 55 analyses have proven this to be a valid approach. Loads for a distribution of connectors can be found by conducting runs using multiple connectors and matching response to the single spring results. Solving the statically indeterminate problem to find the distributed connector loads tends to match the largest value of a multiple spring WAMIT run within 10-20%.

The ships were constrained in sway and yaw. Surge restraint was added to maintain longitudinal as well as lateral relative position. Loads were also evaluated for surge, sway, and yaw restraint.

The points of interest are the connection and collision points. The relative motion between the connection point determines the connector loading. The relative motion between the collision points determines the probability of collision due to wave action.

The connection points were taken at the main deck of the T-ACS and at the fore and aft superstructure. The fore and aft superstructures are the practical limits of the deck "work area". The main deck is taken to be 52 ft above the T-ACS baseline, or 20.42 ft above the waterline. The longitudinal positions are +195.5 and –151.3 ft from the LCG. Relative motions were calculated at these longitudinal positions; the loads were determined at the LCG/LCB. If there are four connection points, they should be evenly spaced between the superstructures, i.e., +195.5, 79.9, -35.7, -151.3 ft from the LCG.

The collision points should be high on the superstructure to be the worst case for potentially rolling into each other. The collision points have the same longitudinal and lateral position as the connection points, but a height of 60 ft above the waterline, approximately the LMSR main deck. Lateral response less than 2.9 ft RMS gives a very low probability of collision, 0.01% in 5 days of exposure to 8-second waves.

4.2.2. Results

The WAMIT transfer functions were combined with the sea spectra to generate a response spectra for each degree of freedom and point location. These response spectra were integrated to give the root mean squared (RMS) value for each degree of freedom.

These RMS values can be multiplied by an appropriate factor to determine significant value or extreme statistics.

The study used the Bretschneider 2-parameter sea spectra. The significant wave heights were the maximum of the NATO sea state bands. The modal period was the most probable period in the open ocean Atlantic on an annual basis. The most probable periods for the Pacific are not much different for these sea states.

Connecting the ships together tends to increase the motions of both ships over the unconnected case. Beam seas tend to be the worst heading as sway, heave, and roll all increase. The increase in sway is because the connection point is not in the same plane (waterplane) as the origin. In Sea State 3, response at origin is still small; heave and sway less than 1 ft RMS; roll and pitch less than 0.5 deg RMS. In Sea State 5, heave is over 1 ft RMS for all headings and up to 3 ft RMS in beam seas. The T-ACS has a roll of 3.15 deg RMS in beam seas. Again the LMSR motions are less than the T-ACS at most headings and beam seas is the worst heading. The relative motion predictions indicate the ships are not in danger of collision due to wave action.

Appendices B-D contain the tables of the response for the body origin and the points of interest. The loads for the single spring connector located at the LCB/LCG are also included. The units are English units throughout the appendices and the values are RMS values.

Appendix B contains the results for the unconnected case. Headings from all sides are shown to verify the headings where the LMSR shields the T-ACS. The response tends to be small due to the large size of the ships and the long natural roll periods. In Sea State 3, surge, sway, and heave are all less than 1.0 ft; roll, pitch, and yaw are all less than 0.5 degrees. Even in SS5, the response is small with roll and pitch approximately 0.5 degrees RMS. Surge is less than 1.0 ft RMS; sway less than 2.0 ft RMS; and heave is less than 4 ft RMS. The response of the LMSR is smaller than the TACS despite being on the weather side due to its large size. The ships have are not in danger of colliding due to wave action, i.e., very low probability.

Appendix C has the results for sway and yaw restraint connection. The ships have zero relative sway and yaw at the connection points. Yaw is the same for both ships at every point on either ship. Sway or lateral response is different depending on vertical location. On at the connection point height is the relative lateral response zero. The change in lateral response is due to unrestrained roll; hence, interest in the collision points.

Appendix D has the results for a connection system that restrains surge as well as sway and yaw. A system where the connectors were pinned or hinged with the hinge axis parallel to the centerline would be an example of this type of system. The failure mode in surge is longitudinal bending rather than buckling for sway or yaw.

Restraining surge does not change the response, other than surge, much from the only sway and yaw restraint case. This is largely because surge is weakly coupled to other modes of motion.

4.3. Comparison of WAMIT & LAMP results

As part of the skin-to-skin hydrodynamic analysis, special zero speed WAMIT runs were conducted and compared to zero speed LAMP results. This was done in order gauge the accuracy of the LAMP data, which have not been validated against any empirical data. The comparison analysis was done by comparing the transfer functions for each of the zero speed cases described in the previous section.

For the most part, the comparison between the WAMIT and LAMP transfer functions were favorable and provide confidence in the accuracy of the LAMP data. The magnitudes of roll, sway force, and yaw moment were similar in magnitude for the zero speed test case described in the previous section.

However, there was a discrepancy in the distribution of the wave energy. This difference in energy could be attributed to comparing a time domain simulation (LAMP) to a frequency domain simulation (WAMIT) or could be the result of slightly different conditions used in each respective study. While the LAMP data is sufficient to assess the feasibility of skin-to-skin operation, further analysis will be required if the numbers are to be used as the basis for design or operational requirements.

5. Ship Control Technologies

This section details concepts, technology, and procedures related to ship control during skin-to-skin operations, including ship stabilization concepts that are designed to reduce motion in order to facilitate cargo transfer. In addition, this section also addresses advanced communication systems and their applicability to skin-to-skin operations, automated approach control, dynamic positioning, vacuum and conventional mooring systems, and advanced fendering concepts.

5.1. Ship Stabilization

As discussed in Section 4, the ships involved in a skin-to-skin operation will experience motion caused by wave action. This motion, primarily pitch and roll, is the main complicating factor in skin-to-skin operations. Large amplitude and/or high frequency motions will place unique requirements on the mooring, fendering, and cargo handling systems of the ships involved. A system that would reduce the magnitude of this motion would therefore reduce the requirements on those associated systems. This study presents four potentially useful systems for consideration: passive flume tanks, an active flume tank system derived from the Ship Roll Stimulator System, rudder roll stabilization, and active fin stabilization.

5.1.1. Passive Flume Tanks

Passive flume tanks are an existing, proven system for roll reduction in rough seas. They have been installed on over 2500 ships since the 1950's.[1] Passive tanks have two features that make them an attractive option. First, their operation is not dependent on ship speed. Second, they are simple and do not require significant amounts of machinery to operate, and are therefore low maintenance. Figure 5.1 shows a diagram of a passive tank system.

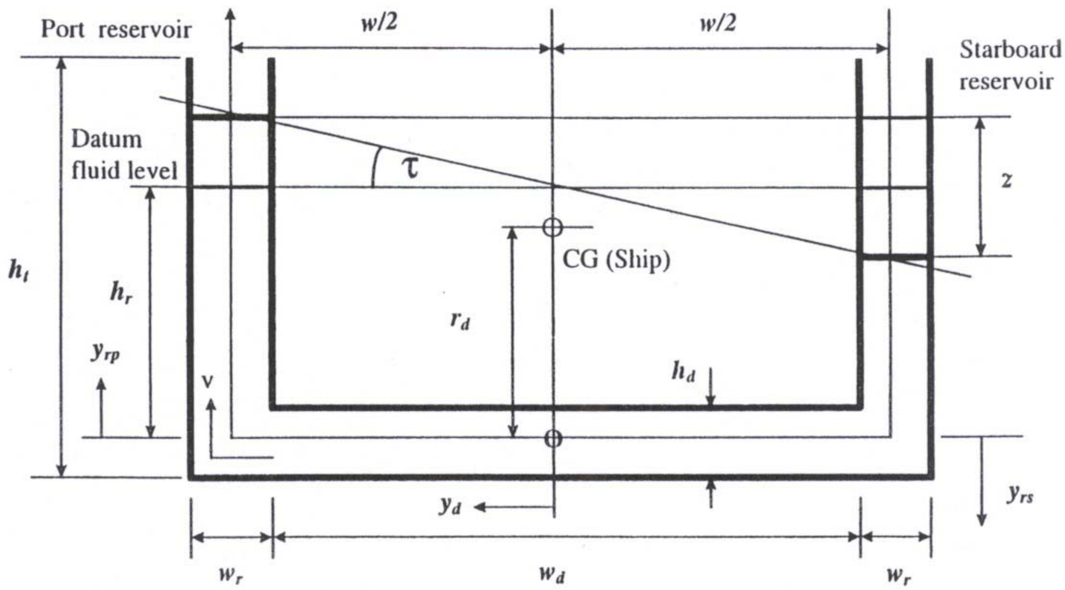


Figure 5.1 – U-Tube Flume Tank Stabilizer

The flume system works by allowing the fluid in the tank, which can be fresh water, sea water, fuel oil, or any other liquid with appropriate viscosity characteristics, to freely flow from one side of the ship to the other as the ship rolls. In a properly designed tank, the flow will be 90 degrees out of phase with the roll of the ship, causing a counteracting roll moment. The effectiveness of passive flume systems can be summarized as providing 75% resonant roll reduction and 40-60% reduction of significant angles at sea, a more accurate indicator of real-life performance.[2] The combination of effective roll reduction with the simple nature of the design of passive flume systems makes them a very attractive option, particularly for ships yet to be built.

5.1.2. Active Flume Tanks w/Ship Roll Stimulator System

5.1.2.1. Introduction

This section assesses the feasibility of the using the Ship Roll Stimulator System (SRSS) as a roll mitigation device. This evaluation is based on the SRSS operation, which went through performance testing at the contractor's facility during August 2001, and aboard the Auxiliary Crane Ship (T-ACS) Flickertail State (T-ACS 5) during November 2001.

The SRSS was developed originally as a modular, active flume system to stimulate roll on T-ACS 5, SS Flickertail State, while pierside or at anchor in calm water. Controlled stimulation of ship roll allows greater flexibility in conducting tests of cranes or other cargo handling enhancements, as well as provide opportunity for more realistic training. The goal for the SRSS was to provide dynamic and repeatable tests on command as opposed to activating the ship and waiting for weather conditions, thereby saving the project time and testing costs. From the performance testing already done with the SRSS, it is concluded that the system has enough capacity to also serve as a roll mitigation system and can be converted with modifications to the stimulator control algorithm.

5.1.2.2. Current State of SRSS/SRMS

The SRSS flume consists of ISO sized components containing 2 sets of 20-foot tanks stacked 3 high, a power unit in the center, and a connecting pipe. One complete unit is shown in Figure 5.2 below. Two identical systems were built and installed aboard T-ACS 5. As of November 2001, the system was fully operation for roll stimulation and was able to obtain approximately +/- 4° roll. The system is scheduled to stimulate the T-

ACS in the first quarter of FY03 for the Advanced Crane Control ATD and will be available for other uses after this event.

Figure 5.2 - Single SRSS Unit Assembly During Shop Testing for Performance Baseline, August 2001

5.1.2.3. Concept Design and Feasibility Issues

For roll mitigation in deep-water applications, the SRSS will need an algorithm to convert the control software from a roll stimulator to a roll mitigator. Some issues that need to be addressed for the application of SRSS for use as a SRMS and determining the best type(s) for the various ships are as follows:

- Ship types and potential for retrofit or new build SRMS installation
- Ship reactions to wave modal periods while underway
- Prediction of hull motion
- SRMS frequency range required to overcome ship motion
- Natural frequencies inherent within current SRSS hardware
- Secondary harmonics induced potential for uncontrolled excitation
- Determining effect of system lag within SRMS control

- Efficiency and power issues when mitigating chaotic wave motion

Collecting this information will allow a thorough analysis of the SRSS' possible effectiveness as a roll mitigator to be performed.

5.1.3. Rudder Roll stabilization

Rudder roll stabilization (RRS) uses quickly applied, short duration rudder movements to produce a moment about the ship's center of roll. On some ships the length of time the rudder force needs to be applied to stabilize the roll is so short that it will not affect station keeping.[3] In these cases, it is possible to use an RRS system.

RRS systems have the potential to reduce roll on the order of 40-50%, similar to passive flume tanks. However, RRS effectiveness is heavily dependent on several factors, including:

- Ship geometry and maneuvering characteristics
- Available rudder rate
- Forward speed
- Heading relative to waves

The ship geometry and maneuvering characteristics are the most important factors to consider when evaluating a ship for potential RRS installation. The rudder must be capable of applying a large roll moment in a short amount of time for the system to be effective. If this is the case, then the ship is a candidate for an RRS system. If not, the system would not be effective and other options would have to be investigated.

If the system is deemed feasible for a given application, the next technical issue to be addressed is the rudder rate. The faster the rudder is able to apply the desired force, the more effective the system can be. The standard rudder rate on most ships is sufficient to achieve some roll reduction, however the percent roll reduction will be greater at higher rates, up to 10 deg/sec. The optimal rudder rate is a design feature that would be specific to the chosen application.

Much like an active fin system, RRS is most effective at speeds above 12 knots. RRS is useful at speeds down to 8 knots, but at much lower effectiveness. In addition, RRS effectiveness is related to the ship heading into the waves. The systems tend to be most effective in beam seas. The potential effectiveness of an RRS system on a ship would have to be studied to determine the total effect of the seas and the RRS system at different headings. The analysis discussed earlier shows that in general head and following seas produce the lowest ship motions, so a RRS system's effectiveness in reducing motions in beam seas may not produce a net gain compared to simply steering the ship to a more desirable wave heading.

The last factor to consider when discussing RRS is that the nature of the system, using the ship's steering mechanism to reduce the roll, necessarily adds a significant variable to the ship's steering characteristics. For this reason, RRS is not normally used when the ship is in close proximity to a hazard, which the accompanying ship in a skin-to-skin operation would certainly be considered. While the steering system will remain effective the operation of the RRS does require a helmsman to be familiar with the system, as it will change the feeling of the ship. Also, while a ship with RRS can effectively maintain it's heading over a given length of time, there will be small perturbations that could cause difficulties in skin-to-skin operations.

As a result of the fore mentioned factors, rudder roll stabilization's limitations make it a poor choice for application in a skin-to-skin scenario. The effectiveness greatly depends on a variety of ship characteristics and operating conditions. Also, the use of the ship's steering mechanism for roll reduction during situations in which an obstacle, in this case another ship, is in close proximity is not advisable. The benefits of RRS are available using other systems that do not pose the same problems.

5.1.4. Active fins

Active fin stabilizers are wing-like appendages on the port and starboard sides of a ship's hull. Active fins have several desirable features. First, they are highly effective in reducing roll in an underway ship. Second, they have a minimal impact on ship weight and volume, especially compared to flume tanks. Figure 5.3 is a descriptive diagram of a fin stabilizer system.

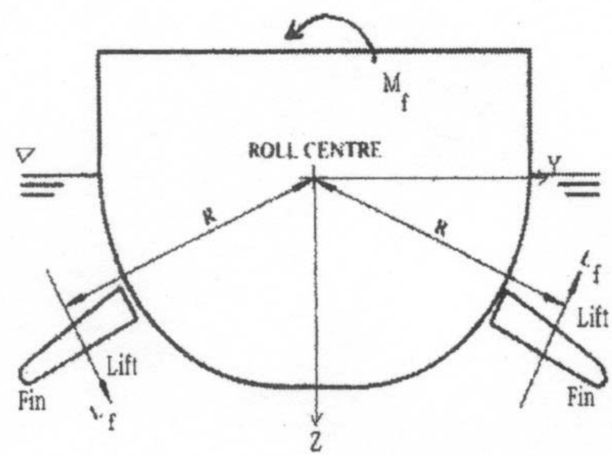


Figure 5.3 – Fin Stabilizers

The fins are designed to produce a lifting force when moving through the water. This force creates a stabilizing roll moment. An optimally designed system is capable of providing a 90% resonant roll reduction, and 60-90% reduction of significant angles at sea.[4] System design parameters include the number, size, shape, and locations of the fins, fixed or retractable fins, and the fin control system. A system would be designed for the specific application and desired performance.

There are significant drawbacks to the active fin system. It is the most expensive stabilizing system. The costs associated with design, construction, and incorporation into the ship's hull is significantly higher than a flume system, as are the maintenance costs associated with the control systems. The major operational limitation for fins is that they are effective only at speeds above 12 knots.[5]

5.2. Free Space Optical Communications & Ranging System

5.2.1. Introduction

The Free Space Optical Communications and Ranging System (FSOCRS) is a system designed to provide a low probability of intercept method of communications between approaching ships, which provides accurate and timely approach course information as well as both voice and high-speed data communications between the ships. This section assesses the feasibility of using this system during skin-to-skin operations. Such a system would be employed in the following phases of a skin-to-skin operation:

a) During the approach, the system would be programmed to report a ship's progress along a predetermined safe optimal approach course, as well as various control settings (engine speeds, rudder angles, thruster controls, etc)

b) During the actual alongside STS-UNREP, data from ship's sensors, including measurements such as line tensions, relative motion and/or position, winch/capstan loads, and control settings, could be monitored such that either individual operators or an intelligent control system could adjust controllable variables (such as winch/capstan tension, vacuum mooring parameters, etc.), thus maintaining a safe and operationally effective skin-to-skin ship configuration. Actual data relative to cargo/fuel transfer could be transferred either in advance or in real time thus leading to enhanced strike up/strike down operations or to improved automatic control of the refueling process

Principal investigators in this effort are:

- Naval Surface Warfare Center, Dahlgren Division, coastal Systems Station, Panama City, FL

- Titan Systems Corp.
- Science & Engineering Associates, Inc

An unsuccessful attempt using narrow beam laser systems was attempted from 2000 into 2002. Although promising, the system was unreliable and relied upon a pan/tilt mechanism for aiming of laser heads on both ships. Each ship was to carry a laser head, which consisted of five separate optical apertures (data send and receive, laser range finder with common send and receive optics, a tracking beacon transmitter and a tracking beacon receiver). Precise alignment of each aperture to all others in the laser head was required for reliable system operation. The complexity of the system contributed to reliability problems and caused excessive projected cost growth.

This new initiative will capitalize on improved high switch rate LEDs to provide a high bandwidth, broad beam, eye safe system. The concept employs an array of wide-angle infrared emitters to overcome the problem of maintaining communications and ranging in high seas. Furthermore, the concept is all solid-state with no moving parts and is based on Commercial Off-The-Shelf (COTS) components. The significant R&D aspects of the project revolve around component selection, component density, operating wavelength selection, application of high-speed power switching circuitry and overall system design appropriate for the environment

5.2.2. Technical Approach

The concept uses an array of light emitting diodes (LED) or laser diodes to transmit both digitized voice and digital data and to determine the range to the other ship. The communication function is relatively simple to achieve with commercially available infrared LEDs. If the LED wavelength is near 940 nm, atmospheric absorption will limit the distance from which the signal could be intercepted by third party sensors, thus contributing to Low Probability of Intercept (LPI). The feasibility of this approach for communication between ships during UNREP was demonstrated in 1977 with a prototype system developed by NOSC called RAPCAP. This system consisted of IR LED emitters and detectors mounted around the perimeter of a hardhat, which was fitted with a microphone and earphones. Wearer's of the units could communicate with each other over distances of 75 yards or more regardless of which direction each was facing or the relative motion of the two ships. By increasing the number of LEDs and arraying them facing the side of the adjacent ship, the FSOCRS will be more robust and have a longer effective range than that demonstrated by the RAPCAP system. A major difference between RAPCAP and the proposed FSOCRS is bandwidth. RAPCAP operated at a carrier frequency of 83 kHz while the FSOCRS goal of a 10 Mbps or greater transfer rate would require at least a 5 Mhz bandwidth. It should be noted that this data rate may be difficult to achieve in bright sunlight based on detector signal-to-noise issues, so even if the LED array is capable of transmitting the full 10 Mbps, the receiver sensor may limit the bit rate. Optimization of wavelength, polarization filtering, multi-wavelength parallel systems (to reduce multipath effects and provide robust error checking), and array size and orientation will be conducted to maximize throughput.

The ranging function for the proposed FSOCRS concept would be accomplished by measuring the time delay of the data pulses that return to the sensor in the emitter array after being retro-reflected from the adjacent ship's array. Although this mimics the approach used in commercial and military laser rangefinders, the use of LEDs rather than laser diodes has not been demonstrated in the UPREP scenario. The key issue here is whether IR LEDs can provide short enough pulses to support the FSOCRS goal of 1 to 3 meter accuracy in the range measurement. A second issue is whether there are COTS LED drivers available that can provide the narrow pulse widths required. The final issue is the availability of a COTS pulse detector with sufficient response time and accuracy to support the ranging function. If technical performance limitations prohibit ranging using this approach either laser diodes or a COTS laser ranging module modified to allow for a wide-angle beam could be used. A key to this overall concept is the desire to stay away from narrow beam optics with their associated alignment and maintenance requirements.

The most important range information is the skin-to-skin distance between the ships. With a single array emitter approach, a single range distance is not sufficient to determine skin-to-skin distance because, if one ship is ahead of the other, the array-to-array distance will be diagonal rather than perpendicular. To resolve this issue, ranging measurements must be made to multiple locations along the adjacent ship's side – placing retro-reflectors at measured points along the ship is one relatively inexpensive and simple way to accomplish this. With the separation distance and the diagonal distance to each array reflector known, the skin-to-skin distance can be derived from fundamental trigonometric relationships.

In conclusion, the Free Space Optical Communications & Ranging System would enable a great deal of information to be monitored and shared between the ships involved in the operation. It is a low-impact system, requiring little in the way of ship modifications and could be used to transfer ship operating data, such as forward speeds, GPS data, control surface information, and other potentially relevant data, perhaps for use in an automated approach and mooring system.

5.2.3. Ship Interfaces

The unsuccessful Laser Phone and Distance Line program did develop both a laptop bridge display and a voice interface unit to allow both headset and sound-powered phone use with the laser hardware. These same units will be used in conjunction with the FSOCRS to allow integration with ships' equipment and crew.

5.2.4. Concept Design and Feasibility Issues

Table 5.1 below presents an abbreviated summary of primary and backup approaches to providing a system to meet the stated objectives. The backup approaches are intended for risk reduction.

Table 5.1 - Alternative Technical Approaches

SYSTEM FUNCTION	PRIMARY APPROACH	FIRST BACKUP APPROACH	SECOND BACKUP APPROACH
Voice Com Link	LED array – 850 nm	Laser diode Array – 940 nm eye safe	
Data Com Link	LED array – 850 nm @ 10 Mbps	Laser diode array – 940 nm eye safe @ 10 Mbps	
Ranging	LED array/retro-reflector – 850 nm, accuracy < ± 3 ft	Laser diode array/retro-reflector – 940 nm, accuracy < ± 1 ft	COTS rangefinder Module – Low-cost @ 940 nm, accuracy <± 3 ft
Voice Com Interface	GFE system developed for L3 hardware	COTS A/D – D/A converter	
Data Interface	GFE system developed for L3 hardware	COTS Ethernet format & hardware – 10 Mbps	COTS RS-232 format & hardware – 2 Mbps

5.3. Automated/Assisted Approach

An automated/assisted approach system would integrate ship sensors, communications, displays, and controls into a system that could either provide direct control of ship systems during the approach phase, or in a less advanced system, be used to present the data required for a safe approach to the crew in an organized, logical, and easy to use manner. An automated system would monitor variables such as ship speed, acceleration, heading, GPS location, and control surface information. This information could be continuously monitored and would allow an automated control system to directly control the ship, or possibly both ships in a coordinated fashion. Control inputs could be made instantly without the delays associated with human controllers.

There is a high likelihood that an automated system would be perceived as unsafe by many in the sailing community. There is a natural and understandable resistance to turning over a safety related function entirely to computers. For this reason it may be best to pursue an assisted approach system. Such a system would monitor the same data as an automated system, but would be designed to present it to the crew in a manner that would give them awareness of all of the variables in a way that is not currently possible. The system could recommend control adjustments, provide predictive information, and give warnings when necessary. A realistic system could incorporate both concepts. A fully automated system could be developed with a control mode that would turn over actual helm control to a crewmember.

Integrated ship control systems, or Integrated Bridge Systems (IBS), are already in existence. Current and future ships that have such systems include:

- DDG-90 and following ships
- LPD-17
- LHD-7&8
- T-AKE
- DD (X)
- LSD-47

The functions of these systems can include radar tracking of contacts (such as the approaching ship in a skin-to-skin operation), auto piloting/heading control, rudder and steering pump control, dynamic adaptation to ship handling characteristics, and propulsion control. These are all existing capabilities that would enable an automated or assisted mooring system to be developed.

5.4. Dynamic Positioning

An important area of consideration in skin-to-skin connected replenishment operations is the ability to control the vessels participating during the cargo transfer. One option for ship control would be the use of a dynamic positioning system to maintain the required stand off distance while underway, perhaps eliminating the need for the ships to be in constant contact with fenders. Dynamic positioning, or DP, can be defined as the use of computer and sensor controlled thrusters along with the ship's main engine and rudder(s) to maintain station or stand off distances. In order for this concept to be valid, several issues must be addressed. Major issues include:

- Operational procedures
- Thrusters/hardware
- Sensors and Control Software

Current dynamic positioning systems are designed for station keeping within a specific watch circle, or radius from a specified location to facilitate operations such as offshore drilling or mine hunting. Based on operational experience, a typical watch circle radius is approximately 5% of the water depth.

For dynamic positioning to be applicable in a skin-to-skin operation, existing procedures must be modified to match the skin-to-skin underway replenishment scenario. All required equipment must be deployed on a single ship so that it can interface with any other vessel that comes alongside in skin-to-skin operations. The vessel without the dynamic positioning system would hold a steady course while the DP equipped ship approaches and maintains the specified stand off distance.

A key to an effective dynamic positioning system is to have sufficiently powerful thrusters with advanced control systems installed on the ship. The large vessels involved with skin-to-skin operations would require thrusters with enough power to operate in a SS5 environment. Some types of thrusters that are capable of proving this control would be retractable azimuthing thrusters, vertical axis propellers, or conventional tunnel (bow and stern) thrusters. It is important to note that any thruster under consideration for DP of a large ship can not be installed on an existing ship due to their size and power requirements. Thruster size and type must be based on envisioned requirements and defined in the design stage of the ship.

Another important component of a dynamic positioning system would be the development of active, feed-forward sensors and the related software to control the system. Feed-forward sensors could be used to anticipate future motions by sampling variables such as wind direction, velocity, wave height, forward speed, current, etc. However, feed-forward sensors have only been applied to wind effects as wave and current characteristics are difficult to measure and wave height is highly transient. Further development would be required in order to apply feed-forward sensing technology.

A DP system would also require the development of an integrated control system to receive the sensor data and manipulate any relevant control systems, such as thrusters, rudders, and main propulsion accordingly to get maintain the desired stand off distance.

5.5. Mooring Systems

The mooring and fendering requirements for skin-to-skin operations are very unique. A greater range of ship motions than in other situations is likely to be present, including the added factor of forward speed. A robust system needs to be developed in order to keep the ships together, maintain a desired separation distance, and prevent collisions. There are a wide variety of currently existing technologies, technologies under development, and conceptual systems that could possibly enable, either alone or in conjunction with others, a safe sea state 5 skin-to-skin mooring operation. This section will discuss: winch technology, conventional mooring line, elastomeric mooring line, and vacuum pad mooring.

5.5.1. Winches & Mooring Line

Modern commercial winches are available in a tremendous range of line pull ratings, line speeds, control designs, and motor designs. Based on the data from the hydrodynamic modeling, current winch technology appears to be adequate for skin-to-skin operations. Conventional mooring line is one of two types: synthetic fiber and wire rope. MIL-HDBK-1026/4A, Mooring Design, lists factors to be considered when determining what type of line to use in a given application. Some of these factors are: break strength, diameter, weight, equipment to be used, stretch/strain properties, dynamic behavior, load sharing between lines, equipment to be used, and cost. Synthetic line has the advantages of ease of handling, and some are able to stretch, in the case of double braided nylon as much as 25% at near breaking strength. Wire rope tends to be more durable.

As discussed previously, the requirements on the systems in a skin-to-skin operation are completely dependent on the properties of the ships involved. The ship arrangements, mooring equipment locations and ratings, relative motions, and forces are all determined by the ships involved. Several issues must be addressed to design a mooring system:

- Required mooring forces
- Relative ship motions
- Mooring equipment locations & ratings
- Ship-to-ship interface

When a specific ship-to-ship scenario is determined, a complete study of the above issues must be performed. Each ship is different and each ship combination will present it's own unique challenges. In general, high speed and line pull winches with fast response times would provide the greatest flexibility and capability for skin-to-skin operations. They would allow variable tensions to be put on the lines, maintaining tension in the lines while the ships move together and high holding power while the ships move apart, and adjustable tensions for different ship combinations and motion conditions. In addition, winch controls could be integrated into a comprehensive monitoring and control system that could allow the winches to provide important motions and distance data during the operation, as well as allow constantly adjustable, automated control of winch operating parameters.

One serious drawback of winch applications on today's ships is that winches are normally mounted on the deck level and fixed in certain locations, usually near the bow and stern of the ship. Depending on the ship-to-ship combination being considered, these locations could be problematic. Figure 5.4 below shows an extreme example of this.

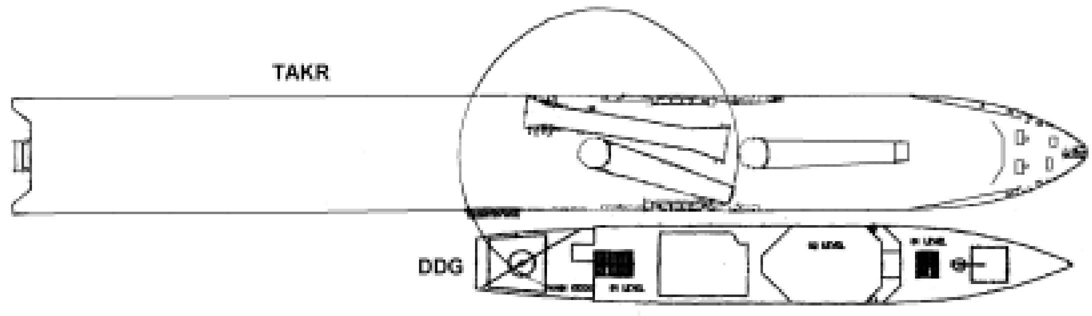


Figure 5.4 – LMSR & DDG-51

LMSR mooring winches are all located at the bow and the stern. In the above scenario, the stern winches would be unable to service the DDG due to the extreme angles present. This leaves the bow winches, which would only be capable of servicing the bow of the DDG. An additional complicating factor is the great difference in freeboard.

Two conceptual approaches for addressing these issues have been developed. First is to develop a mobile mooring winch that could be moved to different locations on the deck in order to compensate for ship size differences. This could be done by using winches mounted on special tracks or rails, or by building different mounting locations that the winch could be moved between with the ship's cargo cranes. The second is to build below deck level winches into the sides of high freeboard ships such as the LMSR, or perhaps more realistically the MPF (F). Such winches could be concealed behind watertight doors at a height closer to the decks of the smaller ships that will likely be involved in skin-to-skin operations. Each of these concepts would enhance the ability of greatly different sized and shape ships to moor skin to skin.

5.5.2. Elastomeric Mooring Line

Straight Moorings International (SMI) is the developer and producer of SeaFlex mooring line, and has provided much of the information presented in this section. The concepts and information presented in this section are proprietary to SMI, and the product name SeaFlex is a registered trademark of SMI.

5.5.2.1. Introduction

The purpose in developing a large "SuperStretch" mooring line is to quickly provide a secure connection between the vessels that will resist surge loading without overstressing the ship or snapping the line.

5.5.2.2. Technical Approach

Integrating an elastomeric painter system into the docking operations would reduce the time required to secure a vessel for loading. The shock-absorbing characteristic of the elastomeric painter allows the wave energy to be dissipated gradually in the mooring lines. However, the docking process of the vessels to each other will require the use of a robust form of fendering that will maintain ship separation, protect the superstructure and absorb large ship motion. The elastomeric painter would provide a secure method for making the initial connection during vessel approach and holding the vessels in position.

5.5.2.2.1. Current State of Elastomeric Rope

Elastomeric ropes have been developed for multiple uses where rope strength combined with the ability to stretch is required. Elastomeric ropes are used as tie down ropes for packages, truck and trailer covers, and are usually referred to as "bungee cord." In the marine environment, elastomeric ropes have been developed for ship moorings, buoy moorings and dock moorings (see Figure 5.5). The rope consists of a strength member core surrounded by a Kevlar braid and covered by an outer abrasion jacket.

Figure 5.5 - Section of SeaFlex™ with Close-up of Cross-Section

For marine applications, the elastomeric rope is specially designed to quickly absorb loading in a stretch, but then locks the energy via the Kevlar braid, and slowly releases it over time. This reduces the spring effect normally found in traditional nylon mooring lines. By reducing the return spring, some damping force is provided. The elastomeric ropes can therefore attenuate a portion of the wave-induced motion by dissipating some of the energy placed on the buoys and vessels as they float in the sea state condition. Another advantage with elastomeric lines is that they reduce the shock forces generated with inelastic line such as cable or chain. Figure 5.6 shows an example of the load reduction provided by SeaFlex™.

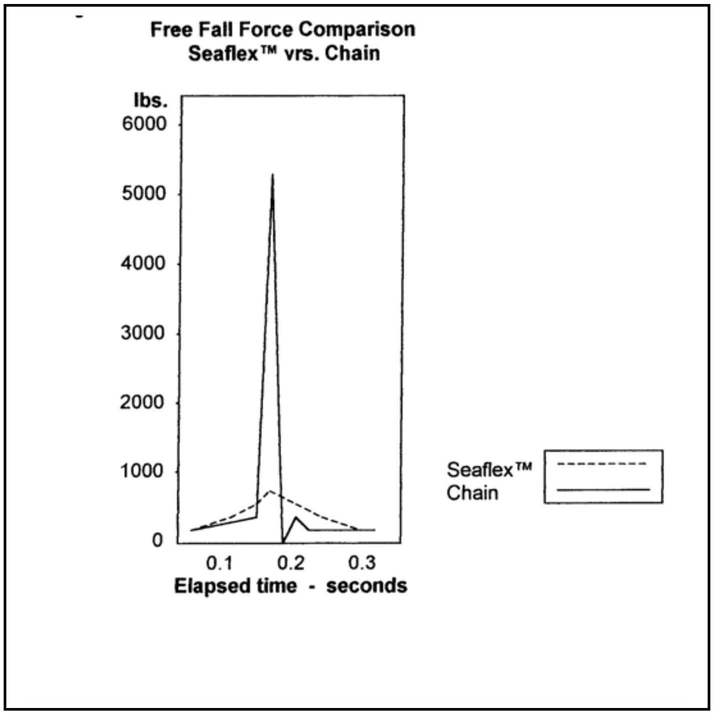


Figure 5.6 - Comparison of Shock Loading Due to Dropped Load

Elastomeric ropes are also used to compensate for tidal conditions where short scope radiuses are required to keep the vessel in a fixed position. Although the elastomeric rope can stretch up to 100% of its original length, the force that can be applied is limited to less than 2,000 pounds as shown in Figure 5.7. To enable forces greater than 2,000 pounds, ropes have been ganged together for the required strength. However, ganging reaches a practical limit of only ten lines as the mechanical hardware to gang more than 10 makes it cost prohibitive. Also, ganging ropes works reasonably well when anchoring under water, but is difficult and dangerous in open-air applications such as ship-to-ship mooring.

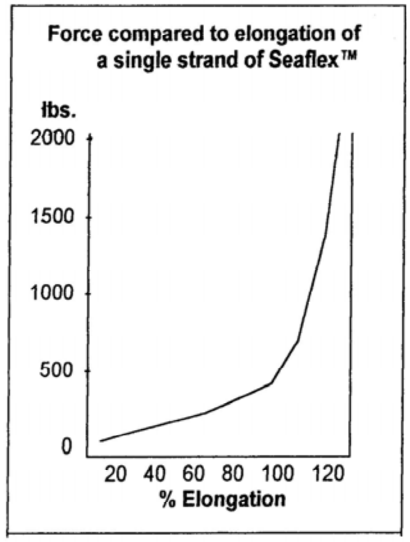


Figure 5.7 - Force Versus Stretch Characteristics

5.5.2.2.2. Concept Design and Feasibility Issues

An ideal elastomeric rope (SuperStretch) would have the following features:

- Provide a single strand rope
- Withstand a force up to 50 tons
- Elongate to 100% of its length
- Work in temperatures of –20 to +40 deg. C
- Resist chaffing in dry open air applications

Using the current commercial technology as a baseline, the development of a large SuperStretch elastomeric rope would require the improved chemical properties of present day polymers as well as a new method for weaving the polymers into a single braided strand to provide the elastic and strength properties desired. Due to the uncertainties in the development, the final elastomeric rope design would be scaled up in dimension to meet the maximum strength requirements. For instance, if a one-inch diameter prototype elastomeric rope showed a strength characteristic of 10 tons when elongated to 100% of its relaxed length, then a two-inch diameter is anticipated to have two times the strength, or 20 Tons. The higher performance of the polymers available and improvements in the weave design will allow a rope to meet the above requirements and can be tailored to meet a number of applications throughout the Navy.

Though the elastomeric rope technology is currently in use for anchoring underwater, scaling the technology for use in ship-to-ship mooring has some unresolved issues. Final size, weight and ease of manual handling for the finished product will not be known until completion of the design. Also, as the intended application is for use in open-air instead of underwater, the amount of heat that can be generated and dissipated through the outer shell is unknown and will be determined greatly by the mechanical and thermal properties of the materials used and the final size of the rope. Abrasion resistance and rope compression issues will follow standard techniques already developed for the outer jacket.

5.5.3. Vacuum Pad Mooring

Mooring Systems, Ltd. (MSL) and SMI jointly prepared the information in this section. MSL is the developer and producer the existing products described in this section. The concepts and information presented in this section are proprietary to SMI.

5.5.3.1. Technical Approach

This evaluation is based on the experience gained from the current products designed and constructed by MSL as well as conceptual designs previously undertaken for the Navy. It is determined that no current automated mooring design in its own right has all of the required attributes to cater for the range of motion and probable dynamics associated with a skin-to-skin application. However, it has been established at a theoretical level, that by combining the characteristics of two previously designed systems, the range of required attributes could be met. It therefore follows that an adaptation and integration of existing designs to produce a new system capable of meeting the needs of the skin-to-skin operation is potentially feasible. Such a new system would require extensive research to ensure operational safety and compatibility with stipulated guidelines.

5.5.3.2. Current State of Vacuum Mooring Technology

Vacuum mooring is a coupling that generates high loads when a vacuum is drawn over a suitable surface area. The technology utilizes computer control to monitor a mechanical substructure producing a mooring interface that in most instances is superior to traditional line securing methods. It is a relatively new but proven technology that combines a variety of engineering disciplines including pneumatics, hydraulics, electrics, control and heavy fabrication techniques.

MSL has developed several system variants that are customized to suit a wide range of operational needs having inherently different mechanical dynamics and holding capacities. In general, MSL's products fall into two categories: port systems and deep sea (or open seaway) systems.

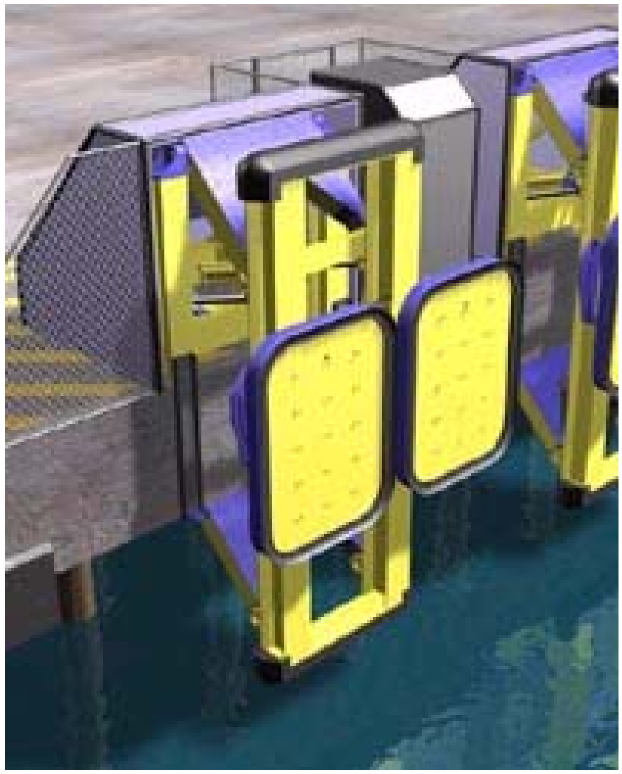
Port based systems have the ability to accommodate normal motions and loads experienced by vessels when alongside a pier. Mooring loads in port are monitored by smart control systems that can anticipate conditions and via associated structure, and damp and dissipate the forces associated with the dynamics of the environment. The operator remains informed at all times of the mooring condition of the vessel.

Seaway mooring systems have been designed at a conceptual level to cope with the magnified conditions of force and movement that are anticipated to occur when two vessels are joined in varying sea states. All mooring variants can either be operated in a fully automatic mode or manually as required.

The vacuum pads provide a direct couple and mooring interface and are specifically designed to cope with maritime applications, whether it be the attachment to the hull of another vessel or to specially designed plates on the shore. The pads have a finite ability to accommodate hull imperfections such as permanent set, marine growth and protruding weld seams.

Dependent on model and required mooring capacity, MSL's vacuum pads have been designed and tested to produce holding forces that range from 10 metric tons to 25 metric Tons. The holding capabilities are dependent on the designed pad area. Examples of automatic vacuum mooring systems are provided in the following sections.

Port Based QuaySailor System



QuaySailor

Name	QuaySailor	
Stage of Development	Full Construction Drawings + Working Prototype	
Holding Capacity	10, 20 and 40 Tonne Units	
Max Angular Displacement		
Roll	4 Degrees	
Pitch	4 Degrees	
Yaw	4 Degrees	
Max Linear Displacement	*Single Unit*	*Stepping Pair*
For-Aft	800mm	Unlimited
Port-Starboard	1000mm	Not Applicable
Vertical	3000mm	Unlimited
Dampening		
For-Aft	Hydraulic/Pneumatic	
Port-Starboard	Hydraulic/Pneumatic	
Vertical	None	
Control System	PLC	

Notes

The QuaySailor Mooring system is a generic quay based mooring solution particularly suited to bulk carriers and RoRo/Pax ferry's operating on fixed routes. This system can be used on new builds or where existing routes operate with minimal or no adaptation to the vessel. When units are paired together in the 'stepping' configuration any vertical or fore-aft range of travel can be achieved. PLC control allows real-time monitoring and feedback of mooring loads and subsequently allows the system to dampen and dissipate any excessive loads induced by environmental conditions.

Ship Based SeaSailor System

SeaSailor

Name	SeaSailor
Stage of Development	Full Construction Drawings
Holding Capacity	10, 20, 40 and 60 Tonne Units
Max Angular Displacement	
Roll	4 Degrees
Pitch	4 Degrees
Yaw	4 Degrees
Max Linear Displacement	
For-Aft	600mm
Port-Starboard	1000mm
Vertical	Unlimited
Dampening	
For-Aft	Rubber Mounting on Quay plates
Port-Starboard	Hydraulic/Pneumatic + Quay plate mounts
Vertical	None
Control System	PLC

Notes

The SeaSailor is a ship based mooring solution running on rails fitted to the hull of a vessel. Each mooring unit attaches to a set of Quay plates on the wharf, which due to their passive rubber mounts provide dampening in the fore-aft direction as well as the port starboard direction. For some applications where fore-aft positioning is required the Quay plates are powered by a Hydraulic/Pneumatic arrangement, which also provides dampening. The SeaSailor systems vertical travel is only limited by the freeboard of the vessel that it is retrofitted to thus removing the necessity to step vertically. By removing the necessity to step unit sizes are often increased and the numbers of units used are reduced.

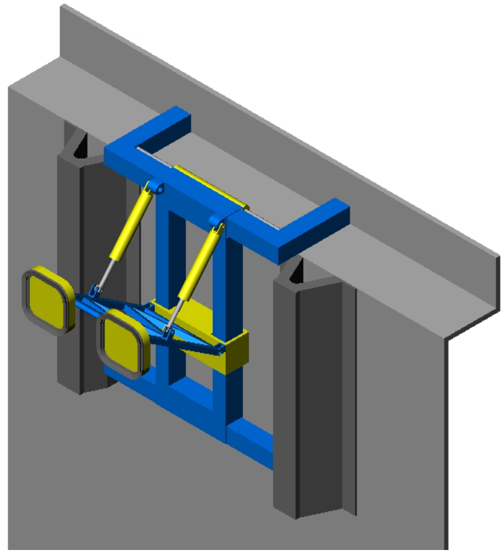
Ship Based IronSailor System

IronSailor

Name	IronSailor
Stage of Development	In Production
Holding Capacity	25 Tonne Units
Max Angular Displacement	
Roll	2 Degrees
Pitch	2 Degrees
Yaw	2 Degrees
Max Linear Displacement	
For-Aft	100mm
Port-Starboard	500mm
Vertical	3000mm
Dampening	
For-Aft	None (Vessels are generally pinned at the link span)
Port-Starboard	Hydraulic/Pneumatic
Vertical	None
Control System	PLC

Notes
The IronSailor is a ship based mooring solution particularly suited to new build vessels due to the necessity for the unit to be housed behind watertight doors. Each mooring unit attaches to a set of Quay plates on the wharf that have a limited range of vertical travel to accommodate tidal and loading variations. The IronSailor does not currently offer a stepping function.

Ship Based Sea-Falcon/Sea-Runner System



Sea-Falcon/Sea-Runner

Name	SeaFalcon/Sea-Runner
Stage of Development	Conceptual
Holding Capacity	25 Tonne Units
Max Angular Displacement	
Roll	4 Degrees
Pitch	4 Degrees
Yaw	4 Degrees
Max Linear Displacement	
For-Aft	600mm
Port-Starboard	1000mm
Vertical	Unlimited
Dampening	
For-Aft	Hydraulic/Pneumatic
Port-Starboard	Hydraulic/Pneumatic
Vertical	None
Control System	PLC

Notes

The SeaFalcon is a conceptual ship based mooring solution for NSWC Dahlgren Coastal System Station. It is merged with Sea-Runner to provide a means to accommodate finite degrees of roll, heave, sway, pitch and yaw. The system has not been designed to cope with dynamic conditions, other than those anticipated (in mooring together two stationary vessels off shore) in lower sea states. Minimum ship alterations are required for retrofit; incorporates passive fendering, near weather deck to prevent superstructure collision and can provide a stepping function. The Sea Falcon was designed with a substantial outreach capability primarily to assist in mooring and positioning a T-ACS to a container ship. Given the T-ACS would likely be anchored in this operation, a suitable (approach) standoff distance pre mooring, would greatly aid skin-to-skin approach of the cargo ship to the T-ACS. Mooring a vessel underway to a stationary vessel can at times, be very difficult and hazardous to ship and crew.

5.5.3.3. Product Overview

The four systems described have been developed either conceptually or to full design. The QuaySailor is currently in the construction stage and IronSailor is in production. As well as these models, MSL also produces scaled variants of its automated mooring systems such as the BargeSailor (a small version of the QuaySailor used for attaching inland barges to ships) and the PaxSailor, a custom built mooring solution for tender crafts and lifeboats attaching to luxury cruise liners. Several other customised concepts have been produced for clients and remain in the development stage.

All mooring systems derive their heritage from the successful and proven, first generation system "IronSailor". MSL's vacuum pads have been tested and rated under the international classification society Det Norske Veritas (DNV) and the units mechanical elements are designed to New Zealand (& Australian) Structural Steel Standard NZS & AUS 3404. Electronic, pneumatic and hydraulic components are specified to their individual ISO standards and are supplied by ISO 9001 accredited suppliers.

5.5.3.4. Conceptual Design

An analysis of existing systems and conceptual systems has been conducted in the production of this report. No single mooring product that has yet been conceptualised or fully designed in its own right to cater for all of the requirements for SS5 skin-to-skin mooring. What has been established at a theoretical level and upon examining the operating characteristics of both the SeaSailor and the SeaFalcon/SeaRunner mooring systems is that both systems combined have in all appearances the necessary attributes needed for a skin-to-skin system.

5.5.3.5. Salient Features of the SeaSailor

The characteristics of the SeaSailor have been summarized previously. The SeaSailor is a sheltered water mooring system that is incorporated into the hull of the ship. Its main feature is the ability to retract the vacuum head and maintain a flush hull form. However, due to the short pivot point behind the vacuum head, its design is limited for use in the benign environment at the pier.

5.5.3.6. Salient Features of the SeaFalcon/SeaRunnner

The SeaFalcon/SeaRunner was first proposed to NSWCDD CSS for application in a Joint Logistics Over-The-Shore (JLOTS) environment to provide a mooring between an anchored T-ACS and containership up to SS3 conditions. The characteristics of this system are covered in the report issued to NAVSEA in late 2001 and the product encompasses the 6 degrees of motion necessary to maintain and control the mooring between the vessels.

However, this design was originally conceived as an add-on to the outer ship hull, thereby requiring minimum modification to the ship being retrofitted. This type of design limits the durability and strength available as well as creates an increased hazard between both ships due to the exposed structure protruding beyond the hull form.

5.5.3.7. Proposed Concept

At a conceptual level, the amalgamated attributes of SeaSailor and Sea Falcon/Runner present opportunities to create a new system to safely moor and endure the dynamics anticipated in a skin-to-skin application. Such a proposal will utilize the following system features:

From SeaSailor

- The extended vertical rail inherent in a SeaSailor design allows a greater latitude to secure vessels of varying freeboard and dimension to a host-mooring vessel.
- The umbilical elements containing control and services to the unit would also be considered in the make-up of a proposed skin-to-skin system.
- The SeaSailor has the structural capacity to resist large moments at the pivot point connecting the pad to operational mechanical elements.

From SeaFalcon/SeaRunner

- The Sea Falcon has a fore and aft travel. The embodiment of this movement capability will be a necessary feature of a new system.
- Passive or active fendering is important. Fendering will provide the only means of dampening the initial hull impact during approach and pre mooring. Provided the vessels are steaming at a controlled speed (i.e. < 5knots) then the impact resultant may in fact be minor. It is prudent therefore to position fendering in predetermined areas according to the needs of the vessels conducting the operation.

The SeaSailor and SeaFalcon were designed to meet a specific and custom mooring need. While both systems embody the ability to cope with 6 degrees of seaway movement, their capacity in this context is limited and unsuitable for a skin-to-skin application. It is of benefit to note that it is entirely possible to extend the movement ranges to allow sufficient freedom without loss of control to cater for skin-to-skin dynamics.

The SeaFalcon has a large operational outreach that is not entirely desirable due to the increased probability of damage in an underway mooring. The SeaSailor's outreach is insufficient. What is required for a skin-to-skin system is an outreach distance in between what has previously been designed. This will mean that the size of the apparatus can be structurally scaled, but remain robust enough to cater for pronounced head loading. It will also be large enough to have an outreach that provides some measure of flexibility during positioning, especially at initial stages and in the transverse plane.

In producing a generation of products that will be suitable for deep-sea application, the following list outlines some other major design elements that will require investigation.

- Dynamic sensing that can determine relative motions between vessels and where possible provide a predictive reaction to seaway motion.
- Control systems that are user friendly that reduce the need for constant monitoring. This may include the Networking of systems on an inter-ship basis by feeding into navigation and propulsion equipment.
- Automation principles that reduce manpower
- Structural considerations as they apply to both hull integrity and system integrity.
- Flexibility in operations given hull design variances versus mooring application. Development of mooring options, both with the hardware and procedures, to allow for various hull shapes.

Fendering properties of the vacuum mooring system and evaluation of additional static fendering integrated into the vacuum system hardware or separately deployed.

5.5.4. Conceptual Design and Feasibility Issues

It will be necessary to establish procedures that relate to approach including speed of approach, impact absorption between hulls and relative positioning, mooring activation and integrity at increased speed, and finally departure considerations.

Operational procedures influence the hydrodynamic forces and energies that a mooring system will accept and inevitably resist during attachment. An analysis of these forces is required before a mooring system can be specified. Current R&D has centered on sheltered water mooring primarily in Australia and Europe, and in seaway conditions not exceeding SS3. The forces induced in such circumstances are generally known and have not posed undue problems in calculations to date. Given SS5 with two free-floating bodies of similar size, the environment is fluid and easily excited. Each body will inevitably exert influence on the other.

The damping characteristics of the mooring units and the effect thereof in this environment is unknown. In general terms it is possible with further research to provide sufficient information to predict what the likely damping and mooring characteristics of a suitable mooring unit would be.

Actual values induced by environmental conditions and hydrodynamic forces will be required before a mooring solution can be designed. In particular, the anticipated range of motion that results when two vessels are moored together. These values will ultimately influence the size of the mooring units and their required dynamic attributes.

The range of vessels potentially involved in skin-to-skin operations will influence the conceptual design of the mooring system. Different sizes will influence the required holding capacity of the mooring system and securing an array of vessels together may provide a wide variance in interaction between hull forms in a seaway.

Differing freeboards will also influence the final design, for example a LMSR has a greater freeboard than a DDG. Subsequently if a DDG were to moor to a LMSR (host) having mooring units at deck level, the units must be arranged in such a manner as to adhere to the *(lower)* shell plating of the DDG. Comparison and analysis between the hull structures and freeboard heights will influence the mooring unit design.

Vacuum pads within MSL's inventory can exert up to 250 Kilo-Newtons (25 tons) of pull of force with just one pad. As a standard procedure, MSL engineers, in the specification of mooring units conduct finite element analysis to determine load effects on hull plating and supporting structure for each ship under consideration to determine the loading distribution needed by the mooring system.

The ultimate structural strength and size of the mooring units is dependent on the distance that the vacuum pads must extend from the hull plating line. The subsequent cantilever that these pads produce on the structure is of particular importance in design. To reduce this moment and unit size it is advantageous to hold the vessels as close as practicable. It must be established just how close is too close and what desired standoff distance in each envisaged mooring application is prudent.

Fore-Aft damping will be a consideration in the tenure of the coupling. It is feasible for the mooring units to conduct a positioning function between vessels in the fore and aft plane.

Given the close proximity of the vessels during approach, it may be determined that a key safety feature is the requirement that one or both vessels have a bow thruster for precise maneuvering while engaging the vacuum mooring. Bow thrusters should also aid in vessel separation at the end of the cargo transfer operation.

5.6. Fendering

5.6.1. Introduction

Clearly any operation where a ship will be in close proximity to another requires fendering to maintain a safe distance and prevent collisions. This section assesses the feasibility of developing a fendering system for the unique situation of skin-to-skin cargo transfer. For the purpose of this study, the vessels considered in the skin-to-skin analysis are: T-ACS, AOE, T-AKE, MPF, MPF (F), LPD-17, DDG-51, LMSR, LSD, and LHA. These vessels are used because they are likely candidates for seabasing operations and information about them is usually available. These vessels and their respective configurations are listed in Table 5.2.

Table 5.2 - Ships and Configuration

Guide Ship	Approach Ship
LMSR (T-AKR 300, USNS Bob Hope)	T-ACS (Flickertail State)
LMSR (T-AKR 300, USNS Bob Hope)	DDG-51 (USS Arleigh Burke)
LMSR (T-AKR 300, USNS Bob Hope)	LSD (USS Pearl Harbor)/(Point Loma)
T-ACS (Flickertail State)	DDG-51 (USS Arleigh Burke)
T-ACS (Flickertail State)	MPF (T-AK 3005, Sgt Mate J Kocak)

The following assumptions were used to guide the feasibility requirements in this section:

- All wire lines and bitts will be adequate to handle the predicted loading.
- Course selection for this report is assumed to be a down wind condition and moving with the seas. The sea state requirement is SS5 operations with moderate seas. No sheer currents are considered. Wind conditions are based on average SS5 wind conditions.
- Speed selection is expected to be from 8 to 10 knots with no controllable pitch systems on the vessels. The lateral separation is expected to be as close as possible with a fender system between the vessels.
- The water depth is assumed to be deep enough to eliminate more pronounced pressure effects.
- Maintaining station alongside the control vessel will be achieved with good seamanship, and the approach vessel will be responsible for primary break away in any situation.
- The Venturi effect is expected to provide a lower pressure, higher speed water area on the sides of the guide vessel. This would affect the control of the approach vessel, and require it to regulate speed, while increasing the difficulty in maintaining station.
- Speed control and quick rudder action is anticipated and the possibility of the vessels breaking away from each other is expected.
- Ice and other cold weather factors are not considered in this analysis since a majority of the operations are to be performed in mild climates.

This section will include a description of the vessel characteristics, ship-to-ship interfaces, loading, and a description and analysis of several fendering concepts.

5.6.2. Vessel Characteristics and Orientation Analysis

Table 5.3 presents the critical vessel characteristics:

Table 5.3 - Critical Ship Characteristics[1]

	Beam (ft)	Depth (ft)	Draft (ft) (Max/Design/Light)	Full Load Displacement (Lt)	Waterline Length (ft)
T-ACS 5, Flickertail State	78	54.5	31.58/30/17.63	26,670/13,530	581.83
T-AKR 300, USNS Bob Hope (LMSR)[4]	105.8	90.3	37/34.67/?	62,096/35,500	890
T-AKR 310, USNS Watson (LMSR)[4]	105.76	90.3	37/34.31/?	62,096/35,500	925
DDG-51, USS Arleigh Burke[2]	66	42.67	31/22/17.67	8,8003/6,691	466
T-AK 3005, Sgt Mate J Kocak (MPF)	105.5	68	33.33/33/14.75	38,960/15320	640
LSD-52, USS Pearl Harbor[3]	84		21/20/?	16,088/11,251	580
LSD, Point Loma	74	40	19.58/18.92/13.08	10,320/6,100	435

1 - All values obtained from Characteristics and Index of MARAD Ship Designs unless noted
2 - NFESC Vessel Program
3 - Naval Vessel Registry
4 - USCG Vessel Exchange Information

An analysis of the candidate vessel combinations was conducted to quantify what impact vessel roll and yaw angles have on defining the required ship separation and fendering dimensions. Figures 5.8 – 5.11 show the configurations listed in Table 5.2 from a profile view and from a top view. The VCG for the analysis is shown be at the waterline but is expected to be below the waterline when fuel loading, cargo loading, and other factors are present. The roll angles are shown with a +/- 20° roll, even though cargo transfer would only take place at a much lower roll angle. This high roll angle was chosen to provide a very large factor of safety. The modeling studies predict motions rarely exceeding 3°, so choosing 20° makes the recommendations for minimum separation distance very conservative. The yaw angle is shown as 2 degrees and indicates the potential for impact. The vessel diagrams shown are presented with squared corners, which provides an additional factor of safety for compensation in roll conditions since the vessels have rounded hulls and deckhouses that are stepped inward towards the centerline.

The required separation distance is defined to offer an approximation of the minimum sizing of the necessary fendering to protect the vessel from impact. For this roll condition, a 20' separation distance is sufficient for all cases except the LMSR-LSD combination, which requires a 30-foot separation. The LSD will require an independent analysis to determine impact points. Table 5.4 lists the minimum separation distances for each configuration. For closer mooring, an analysis would have to be performed to account for the specific mooring configuration. The most likely interferences would take place at: the aft deckhouse, stern ramp, and pedestal cranes for the T-AKR 300; the forward and aft deckhouse, pedestal crane boom house structure and boom stowage locations for the T-ACS 4; the deckhouse profile for the DDG 51; the crane house and boom stowage locations on the T-AK 3005; and the profile of the deckhouse on the LSD 52.

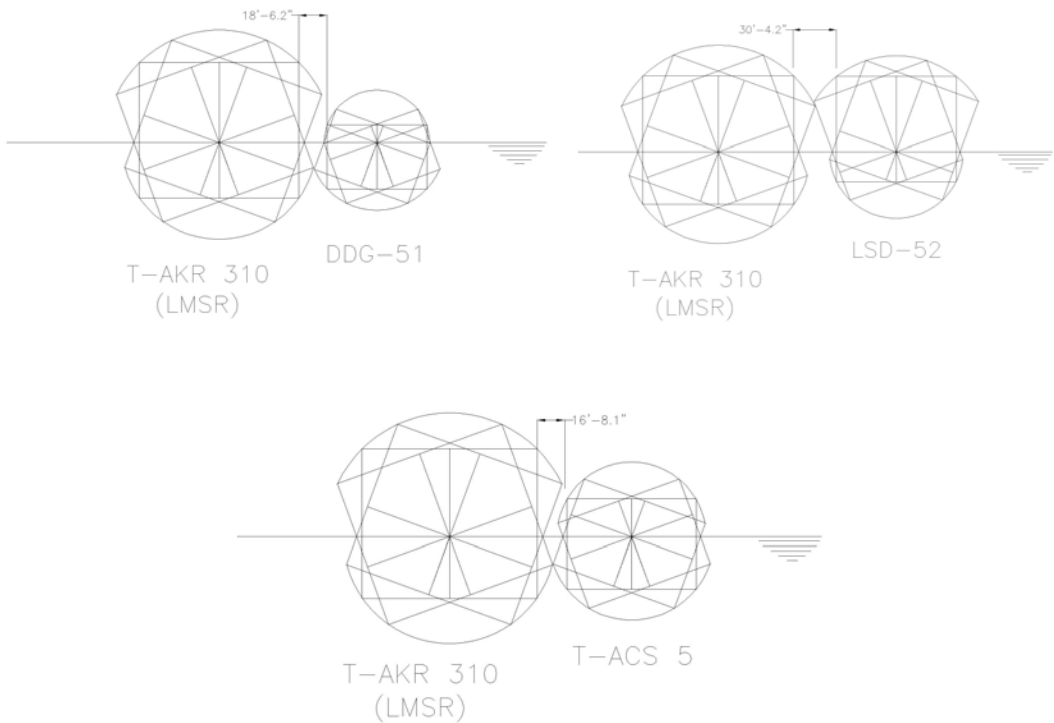


Figure 5.8 – LMSR Moorings

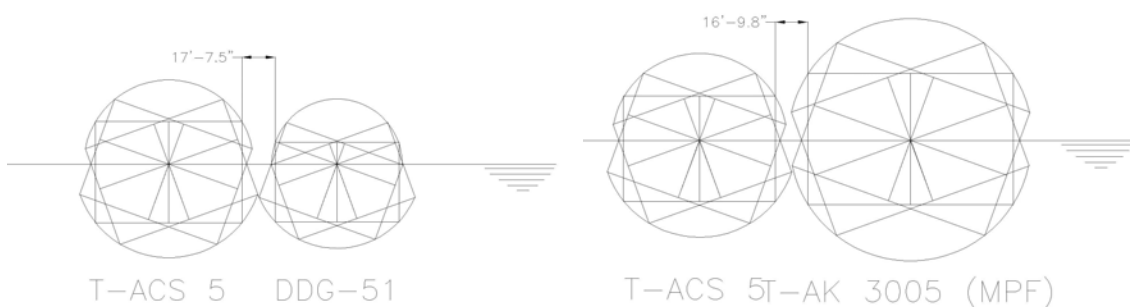


Figure 5.9 – T-ACS Moorings

Table 5.4 – Minimum Separation Distances for 20° Roll

LMSR – T-ACS	16’ 8.1”
LMSR – LSD	30’ 4.2”
LMSR – DDG	18’ 6.2”
T-ACS – DDG	17’ 7.5”
T-ACS - MPF	16’ 9.8”

To examine the effect of yaw angles on each configuration, scaled drawings were constructed that depict the vessels with the bow separation from the roll analysis and simultaneous 2 degree yaw angle on the approach ship. Figures 5.10 and 5.11 show the arrangement of each vessel combination. Table 5.5 lists the resulting separation at the bow and stern of each configuration. This table shows that without fendering, the bow separations are less than 20 feet with the LMSR/DDG and T-ACS/DDG configurations having the smallest separation.

During berthing, the bows of all of the ships fall within the 20-foot fender separation area, while the aft of the ship is greater than 20 feet. This means that the front fender will be fully engaged before the aft fender is engaged at all. If the energy absorption of the fendering system depends on fender compression, the lateral separation produced by the compression needs to be considered. Compression across the forward, midships and aft fendering has to be included in the energy absorption distribution analysis and determination of the reduced capable berthing angle of the vessels. Commercial fendering is estimated to compress to 50 percent and would produce a 10-foot lateral distance for compression. This 10 feet, over the length of the DDG and T-ACS vessels produce an approximate 1.25 and 1-degree maximum berthing angle respectively.

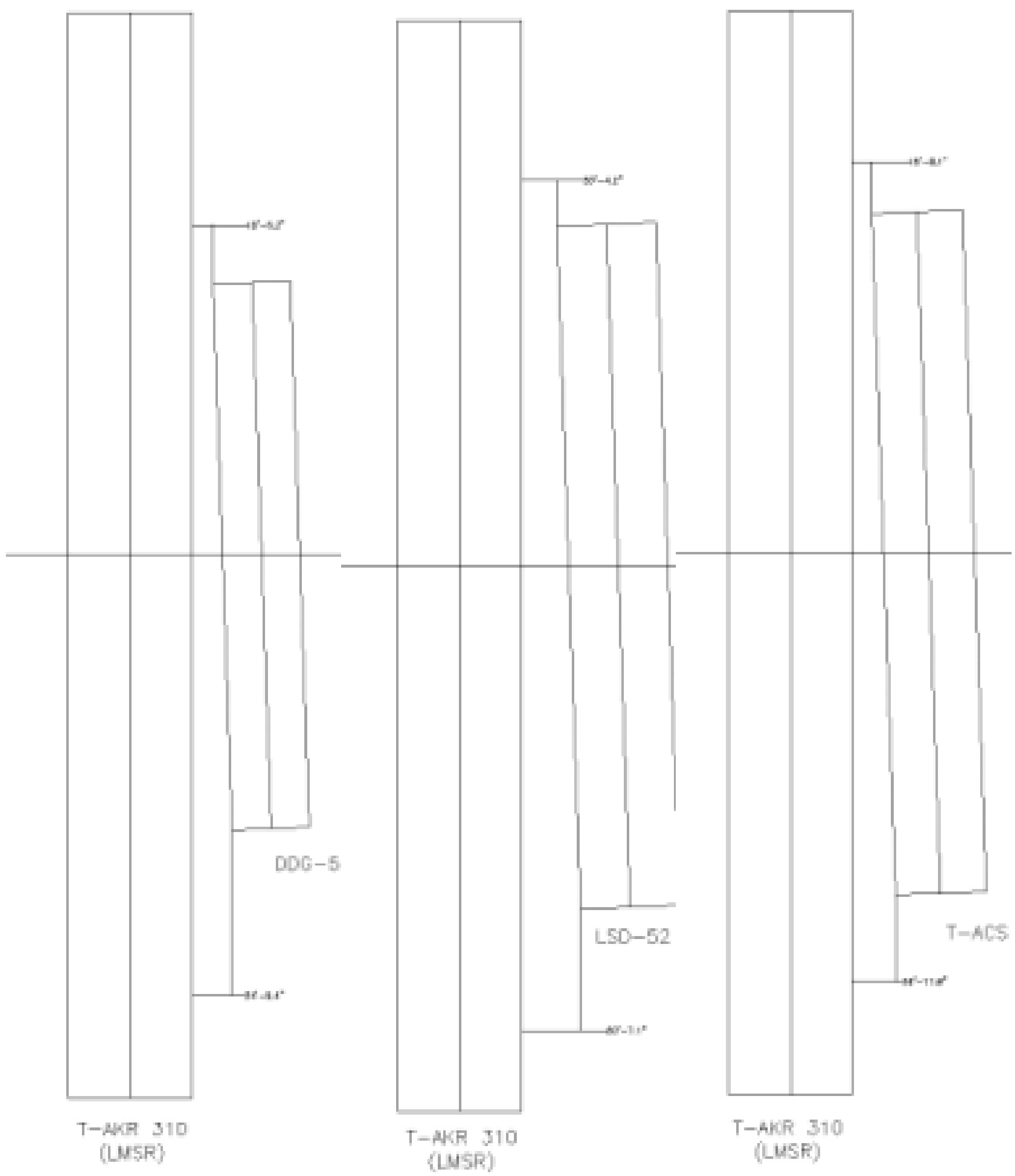


Figure 5.10 – LMSR Moorings 2 Degrees Yaw Angle

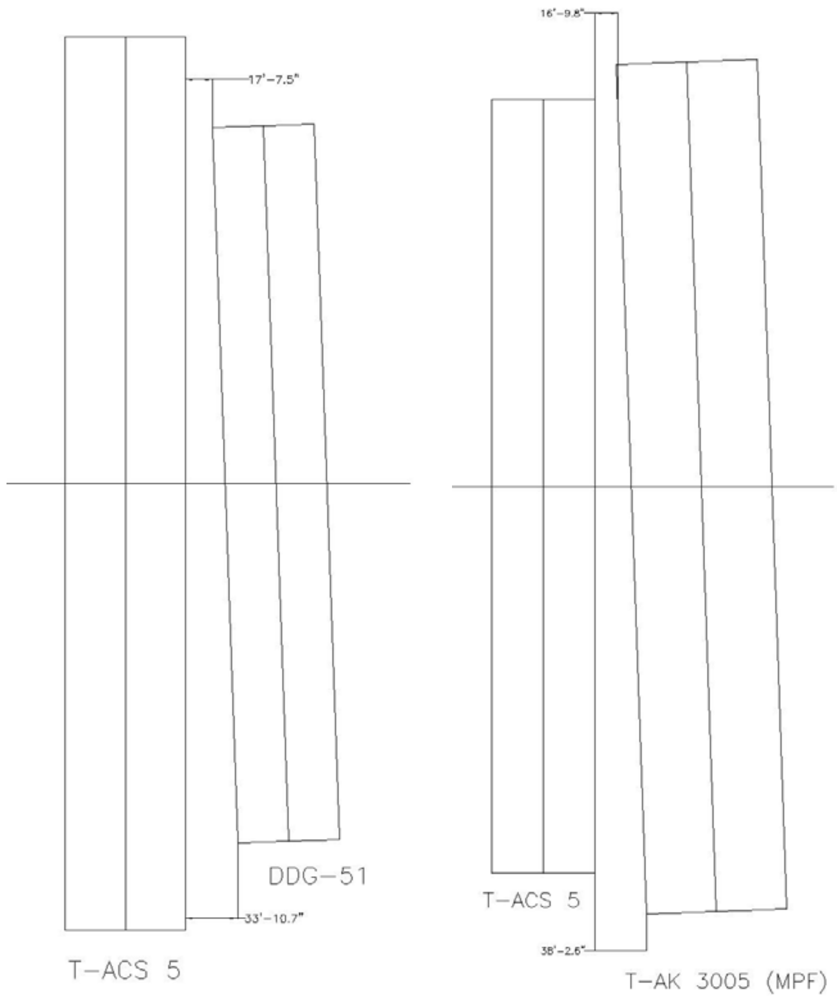


Figure 5.11 – T-ACS Moorings 2 Degrees Yaw Angle

Table 5.5 – Potential Bow and Stern Separation at 2 Deg Yaw

	Bow	Stern
LMSR – T-ACS	16' 8.1"	38' 11.8"
LMSR – LSD	30' 4.2"	50' 7.1"
LMSR – DDG	18' 6.2"	34' 9.4"
T-ACS – DDG	17' 7.5"	33' 10.7"
T-ACS - MPF	16' 9.8"	38' 2.6"

Impact areas are assumed to be on the forward and aft 1/3rd of the vessel and can be deck to deck impacts, deckhouse to deckhouse impacts, or chine to chine impacts, and would require a minimum of two fendering locations for protection. The impact on the forward 1/3rd would be due to oncoming sea motions, steering motions, and wind direction changes. Impact due to steerage would be on the aft 1/3rd due to the rudder control. If the rudder control is in that aft 1/3rd, the front of the vessel has a range of motion that it experiences prior to the rudder enacting a heading change. A change in the rudder would move the rear of the vessel laterally to change direction of the vessel heading. Therefore, if the fender is located at the forward and aftermost areas of the smaller ship, and centered about the LCG of the guide ship, it will minimize the roll and angle induced by steering control and sea motions.

Hull pressure is an important factor to include as part of the impact analysis. Each impact will form a pressure or force on the area contacted. Each ship has a specific hull thickness along with various structures and supports behind the hull plating. Some ships have thicker hull plating for military or ice breaking efforts, while others have thinner hull plating due to the commercial nature of its design. Table 5.6 presents the hull contact pressure as reported in MIL-HDBK-1025. These values are not reported for specific ships, but fendering design should use hull pressures in the 12-17 psi range. It is assumed that these ratings are for open areas of hull contact, meaning that no structure is assumed to be behind the plating. In the places where structure (specifically, the bulkhead structure) is located, significantly higher loading is assumed. A structural analysis based on placement and berthing energies is required upon fendering concept selection.

Table 5.6 - Hull Contact Pressure

Hull Contact Pressures for Various	
Ship	**Hull Contract Pressure (Kip/ft2)**
DDG 993	1.34
DDG 37	4.3
DDG 2	4.6
LSD 36	2.6
Amphibious Ships	1.7-2.4

5.6.3. Estimated Loading

Tables 5.7 - 5.13 summarize the berthing energy from 0.5 to 2 degrees, and from 8 to 10 knots. This speed range is chosen because it covers the minimum speed required to maintain ship control with rudders.

Table 5.7 - TACS - 5 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	11.20	44.79	100.80	179.26
8.5	12.64	50.56	113.79	202.37
9.0	14.17	56.69	127.57	226.88
9.5	15.79	63.16	142.14	252.79
10.0	17.49	69.98	157.50	280.10

Table 5.8 - T-AKR 300 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	24.48	97.92	220.39	391.94
8.5	27.63	110.55	248.80	442.46
9.0	30.98	123.94	278.93	496.05
9.5	34.52	138.09	310.78	552.69
10.0	38.25	153.01	344.35	612.40

Table 5.9 - T-AKR 310 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	24.48	97.94	220.42	392.00
8.5	27.64	110.56	248.83	442.53
9.0	30.98	123.96	278.97	496.12
9.5	34.52	138.11	310.83	552.78
10.0	38.25	153.03	344.41	612.50

Table 5.10 - DDG51 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	3.96	15.84	35.65	63.41
8.5	4.47	17.88	40.25	71.58
9.0	5.01	20.05	45.13	80.25
9.5	5.58	22.34	50.28	89.42
10.0	6.19	24.75	55.71	99.08

Table 5.11 - T-AK 3005 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	14.75	59.00	132.77	236.13
8.5	16.65	66.60	149.89	266.57
9.0	18.66	74.67	168.04	298.85
9.5	20.80	83.19	187.23	332.98
10.0	23.04	92.18	207.46	368.95

Table 5.12 - LSD 52 (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	5.60	22.39	50.40	89.63
8.5	6.32	25.28	56.89	101.18
9.0	7.08	28.34	63.78	113.43
9.5	7.89	31.58	71.07	126.39
10.0	8.75	34.99	78.75	140.04

Table 5.13 - LSD (loaded) Horizontal Components

Knots	@ 0.5 Deg (Kip-ft)	@ 1.0 Deg (Kip-ft)	@ 1.5 Deg (Kip-ft)	@ 2.0 Deg (Kip-ft)
8.0	3.66	14.64	32.96	58.61
8.5	4.13	16.53	37.21	66.17
9.0	4.63	18.53	41.71	74.18
9.5	5.16	20.65	46.48	82.65
10.0	5.72	22.88	51.50	91.58

To develop design criteria for the fender, a maximum loading of 612.50 ft-kips should be used. The berthing calculations assume the individual vessel is berthing against a solid surface, where as in the skin-to-skin scenario the berthing energy will be a sum of the components of the berthing energies of both vessels. The maximum energy is assumed to be the energy of the larger vessel at the differential angle between the vessels and the differential speed. Roll loading and other loading should be assumed to be in addition to berthing energy.

Energies produced by the mooring lines and developed headings are assumed to be smaller than the berthing energy and transmitted through fendering and mooring lines. Mooring lines can cause resonate loading and will have to be selected based on the spring factor of the fendering and sea conditions.

5.6.4. Fendering System Descriptions

Several conceptual approaches, all with the ability to provide the required fendering and standoff distance, have been developed for discussion. Six generic categories are defined and evaluated, and listed in Table 5.14.

Table 5.14 - Fendering Concepts

Category	Concept	Operating Zone
	Pneumatic	
Suspended	Composite Box	
	Extension Arm	
	Articulated Arm	
Suspended Articulated	Parallel	Air
	Fendering Sled	
Surface Towed	SWATH Fendering Sled	Water Surface
Submerged Towed	Vertical Wing	
Waterjet	Waterjet	
Static Mixer	Static Mixer	Below Surface

Each of the ten concepts is described below. For this study, each concept is to be repeated at two longitudinal locations on the guide ship, while the concept descriptions will address only one such station.

5.6.4.1. Suspended Fenders

The suspended fendering concept involves suspending large elements horizontally from the deck fittings such that the fenders do not enter the water, but are located as close to the waterline (or peak waves) as possible. Two general approaches to this concept are defined; pneumatic and composite box fenders.

The first is a set of three connected pneumatic fenders (Yokohama) measuring 11’ 6” in diameter by 26’ 3” in length. Each unit weighs 20,400 pounds. To provide the required 20-foot ship separation, three pneumatic fenders would be arranged as shown in Figure 5.11. The ends of the three would be married via pendant wires. Presently, the T-

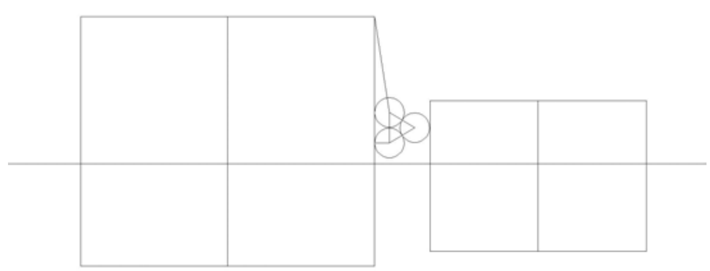


Figure 5.11 – Suspended Pneumatic Fendering

ACS uses single Yokohama fenders for zero speed moorings with containerships. A more detailed analysis is necessary to determine if this single fender arrangement would be suitable for skin-to-skin operations. Additionally, low pressure fenders such as those manufactured by Dunlop may be suitable. These fenders have the advantage of large size, up to approximately 15' diameter and 100' length, low weight, under 10,000 pounds, and very low hull pressures due to the 1 psi inflation pressure. An additional advantage is that these fenders are collapsible and can be stored in almost any location, and handled with standard small cargo handling devices. Again, additional research is needed to determine if these low pressure fenders, either alone or in concert with Yokohama fenders, would be a suitable system.

The second approach to the suspended fender concept is the design and production of a fendering box structure that has ISO standard connections, which provide for standard storage but allow assembly of two or more units to provide the required 20-foot standoff distance. Both inboard and outboard surfaces would be equipped with standard Commercial Off The Shelf (COTS) fendering elements. The use of composite structural materials and appropriate sizing offer weight reduction advantages over the pneumatic

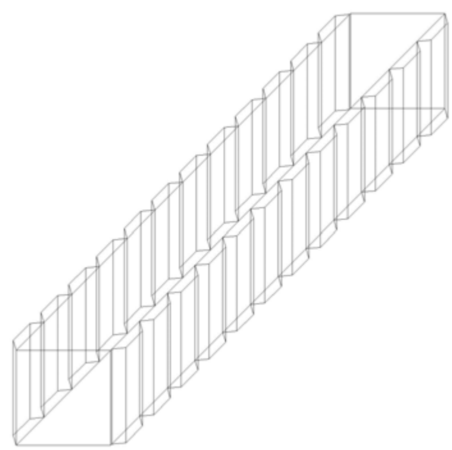


Figure 5.12 Suspended Composite Box Fender

approach. The suspension system would need to be developed to handle and control the box fender assembly. Figure 5.12 shows an example of a suspended box fender.

5.6.4.2. Suspended Articulated Fenders

The suspended articulated fender concept is similar to the suspended fenders, but with an added articulated feature. To provide adjustability or collapsibility, three such arrangements are identified.

The first articulated concept utilizes an extension arm approach. The concept uses a standard fender cushion and pivoting links to form a modular sub-assembly that can be configured to provide a range of end-angle alignments to compensate for various ship-hull geometries and orientations. Several modules can be joined together to provide the required lateral separation.

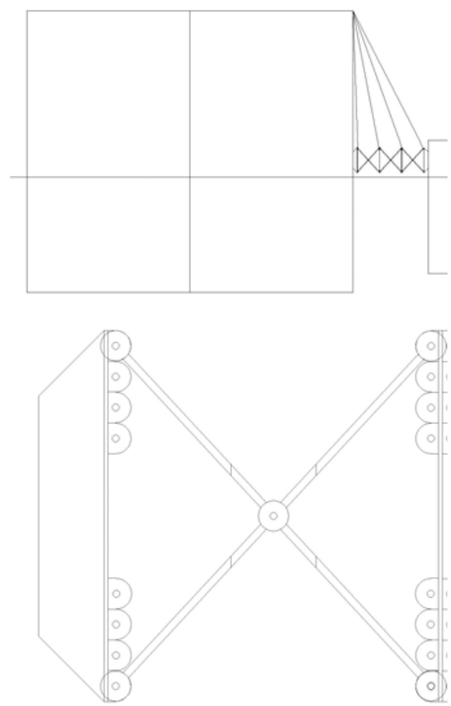


Figure 5.13 – Extension Arm

Figure 5.13 shows the individual module and an arrangement of a three-module installation.

The second articulated concept, Figure 5.14, is a folding arm arrangement of an array of COTS fender elements, with two long and one short link. This geometry can be used in several ways. When it is suspended from one end, it can be used as a standard fender for zero velocity mooring with a portion suspended into the water. A second application is to fold over the arrangement to provide an added standoff distance. In this arrangement, the energy absorption capability is increased due to the parallel arrangement of fender cushions. A third function could be provided in an "inverted V" attitude if the link pins are equipped with coil springs or rotary clutches to provide motion damping.

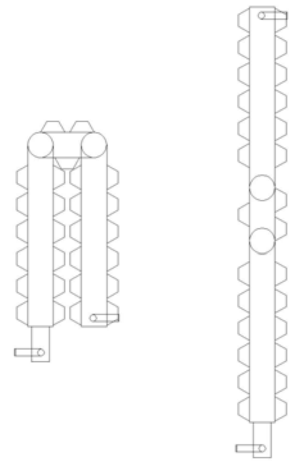


Figure 5.14 – Articulated Arm

The third articulated concept, Figure 5.15, consists of a pair of suspended box structures that are joined by rotating linkage arms. When in transit, both box structures would be fastened in tandem, parallel to the longitudinal axis of the guide ship. For operational use, one box structure would be pivoted outward on its linkage to increase the standoff distance and serve as the ship-to-ship fendering system. The linkage pivot joints could be equalized with springs or clutches to provide motion dampening. For finer tuning and control, the joints could be equipped with force

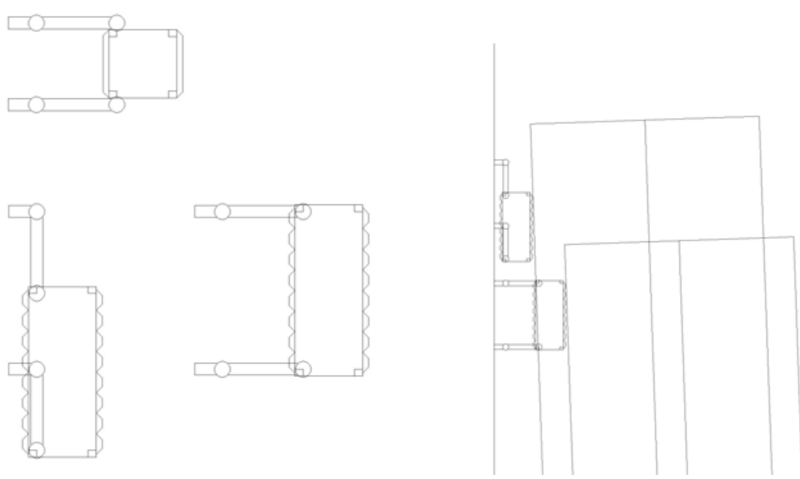


Figure 5.15 – Parallel Fender

control elements that employ Magneto-Rheological (MR) fluids for tunable damping characteristics.

5.6.4.3. Surface Towed Fendering Sled

Two configurations of a fared fendering sled at the air-water interface are defined below:

The first is a buoyant box fender arrangement with fender cushions on both inboard and outboard panels. The interior volume is allowed to flood with water providing a highly damped response in both pitch and roll. The bow members could have flow through points to reduce its towing load. A towing bridle is used to augment the towing stability. Interior stiffener bulkheads also have flow through points. Figure 5.16 shows its general arrangement.

The second towed sled fender arrangement utilizes the Small Waterplane Area Twin Hull (SWATH) technology. The SWATH geometry provides catamaran-style geometry that is buoyant when placed in the water at zero tow speed and supports the deck structure above the water surface. The twin hulls are low-drag, fully submerged torpedo shapes that are connected to the above-water decking that joins the twin hulls.

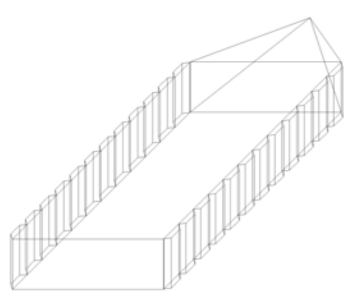


Figure 5.16 – Fendered Sled

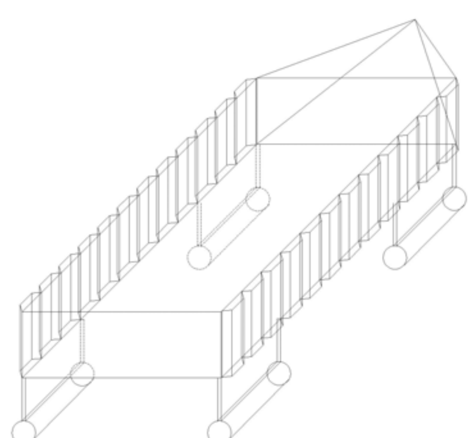


Figure 5.17 – SWATH Fender Sled

The vertical struts that join each pontoon to the decking has low drag shape with small surface area at the air/water interface. The main benefit of the SWATH design is that it provides good hydrodynamic towing characteristics that are very insensitive to wave and sea turbulence, while providing low resistive towing forces. Figure 5.17 illustrates the conceptual arrangement for a SWATH-style towed fender sled.

5.6.4.4. Submerged Towed Fenders

To provide fendering below the water surface for several of the deep draft vessels, a submerged arrangement may be necessary. Since the current study requires the participating vessel to maintain forward speed in the 8-16 knot range, any submerged fender must meet the hydrodynamic requirements of a towed body. The first problem with towing physical objects in the water is hydrodynamic drag, or resistance, which tries to expel the object up and out of the water. To overcome this, a second downward lifting force must be provided, either by weight, hydrodynamic lift, or a combination of both. Most often, the combination is employed while using high lift to drag streamlined foils to provide most of the downward force, while adding the minimum necessary drag associated with the physical size requirements. A small amount of weight is used to control launching and retrieving, and to provide adequate stability at both low-speed and full-speed conditions. Figure 5.18 shows a general arrangement of such a configuration that uses several lifting vanes to simultaneously provide the required downward force together with the needed strength and energy absorbing characteristics. Each wing is fabricated using COTS fender elements and cushions, with modifications to provide them with low drag cross-sectional shapes. The collective strength of all of the streamlined elements must be equal to or greater than the required fendering strength.

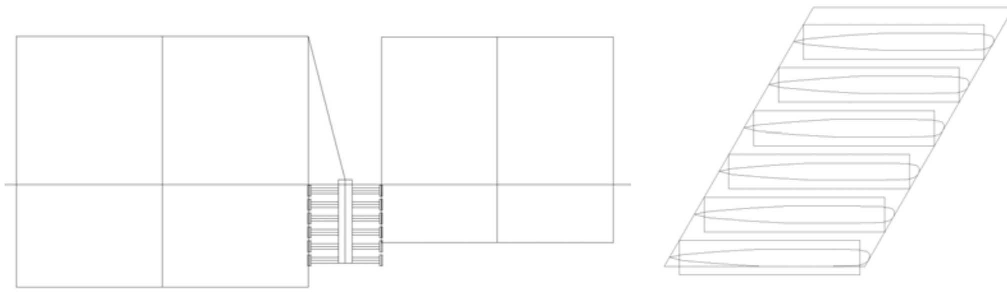


Figure 5.18 - Vertical Wing Fender

5.6.4.5. Waterjet Repulsion

The use of waterjets is often used to direct and control physical objects under and on the water surface. A concept to use waterjets to provide resistance and motion damping between two adjacent ship hulls is defined for the feasibility study. Several arrangements were identified that might be adaptable to the hull structure of a guide ship. However, there is one clear configuration that appears to be the most obvious approach. It is understood that the jets would require one or more supply pumps for activation. But due to the large size and mass of most candidate ship configurations, it is obvious that a large number of jets, high flow velocities, and high flow volumes would be needed to create the force levels required to retard the lateral motion of the approach ship. This suggests that a large network of piping and jets with attachment hardware is also necessary.

5.6.4.6. Static Mixer Repulsion

Another concept for controlling and diverting water flow is identified, and would employ submerged vanes along the hull of the guide ship that would redirect longitudinally flowing water to a lateral flow condition, thus diverting the water flow at the approach ship. The general concept is to create a condition that is the reverse of the suction Venturi effect that is produced when two hulls are in close proximity to each other while moving forward. The vanes have to be moveable and controllable to be able to produce the force needed for the repulsion task, but retractable when normal operations are warranted. Two added issues that need further examination are the steering effect if the vanes are employed on a single side of the guide ship, and the effect on forward speed caused by the anticipated high hydrodynamic speed caused by the anticipated high hydrodynamic drag forces on the vanes, and hence on the hull of the guide ship.

5.6.5. Concept Evaluations

Each of the concepts that are presented earlier are examined individually and judged based on its ability to perform, while taking into consideration the previously defined factors. Following is a summary of the ten concept examples in the same order of presentation as in Table 5.14.

5.6.5.1. Suspended Fenders

The evaluation of both the suspended pneumatic fender concept and the suspended composite box fender concept gave similar results, and therefore are discussed together. Pneumatic fenders are known commodities, and have been employed in shipboard applications for several years. Their normal application within the Navy is for use in stationary mooring situations (zero forward velocity situations). In the current study, one operational requirement is that both the approach and guide vessels maintain a continuous transit speed of 8-16 knots. But since both the pneumatic fender concept and the composite box fender concept cannot be used in or on the water at these speeds, they must be suspended above the water. While it is feasible to hang either the pneumatic fender or composite box fender over the side of the guide ship, each represents a significant weight penalty.

Each Yokohama fender unit weighs approximately 20,000 pounds. Since the pneumatic fender concept is defined as utilizing three such units to meet the lateral separation requirement, this brings each fender station up to approximately 60,000 pounds. While the total number of stations has not been determined, it is assumed that there will be at least two, and more likely three, fender stations. This adds 120,000-180,000 pounds of pneumatic fender units hung over one side of the ship, which would require some amount of counterbalancing weight on the opposite side to maintain level ship stability. This problem could be minimized if it is determined that single Yokohama fenders, or a combination of the light weight, low pressure fenders and Yokohama fenders would be suitable.

Since the composite box fender arrangement is only conceptual at this time, it is probable that a box fender structure can be produced with a lighter arrangement weight than that of the pneumatic fenders. However, if even a 50% weight savings is possible, the resultant weight level still leaves a difficult situation, as the center of gravity of the composite box fender units would be outboard approximately 10 feet from the hull and would create a large moment. Additional difficulties exist in the rigging and handling of the composite box fenders.

5.6.5.2. Suspended Articulated Fenders

The second category for discussion contains three candidate examples for this approach. They are: Extension Arm Fenders, Folding Arm Fenders, and Parallel Fenders.

All three of these examples intend to meet the lateral distance requirement with considerably less weight than the suspended fenders. Each has some degree of articulation, thus increasing their utility and adaptability.

A benefit of the Extension Arm Fender concept is that it is modularized and can be handled and stowed in smaller pieces than the full assembly. However, there exist significant rigging and control line issues which present potential problems.

Both the Folding Arm Fender and the Parallel Fender concepts are configured in a permanent mounting arrangement.

All three of the articulated concept examples present severe design problems due to their pivot joints and linkages.

Any improvement in air weight for the three concepts can only be evaluated after preliminary design efforts have commenced to quantify the element sizes, materials, and resulting weights.

5.6.5.3. Surface Towed Fendering Sled

The third category contains two concepts that provide a fender system that operates at the air/water interface: the buoyant box-fender sled, and the SWATH box-fender sled. They both will be designed with sufficient buoyancy to float freely and independently of the guide ship. Mooring/towing lines will be used to help maintain station over the 8-16 knot speed range.

An advantage of these concepts over the suspended fender concepts is that their weight does not present a full time burden to the host ship, as weight is only an issue during launch and retrieval operations.

The first concept is a simple surface-riding sled that is constructed of durable composite material and conventional fender cushions. While this concept eliminates the weight difficulties of the suspended fender concepts, it has standard sea-keeping properties, which means its performance will deteriorate as sea state conditions increase.

The SWATH fender sled employs technology that has been developed specifically to address rough sea and surface conditions. The elements that provide the major portions of its buoyancy are streamlined, low drag, fully submerged pontoons in a catamaran configuration, which provides excellent roll stability. The structural decking that spans above the twin pontoons is connected by vertical struts in a manner that holds the deck above the water surface where it is free of all water contact. The vertical struts are low-drag and low-buoyancy surface piercing foils. For added rigidity, submerged lateral foils can be designed to join the pontoons. This arrangement can provide added lift due to the 8-16 knot tow speed. The general construction of the SWATH fender would be composite materials for their low maintenance and durability in the marine environment.

5.6.5.4. Submerged Towed Fender

The submerged towed fender concepts avoid the difficulty of towing in the wave zone by being totally submerged; however the combination of tow speed and lateral separation produce large amounts of hydrodynamic drag.

A submerged towed fender system can best be implemented by minimizing the overall frontal area of the towed system, while producing sufficient lift, or 'down force', to prevent the fender from flying to the surface.

The winged submerged towed fender configuration represents a feasible approach to resolving the technical issues associated with towing such a device. However, the energy absorbing requirement will have to be met by stressing the foil-shaped lifting elements, which might cause the fender to have variable towing attitudes and/or behavior. Long-term effects of repeated stressing might render the unit unusable. Tow cable loading is expected to be quite high, and in the event of tow line failure, the unit will be lost due to its weight in water and insufficient buoyancy to keep itself afloat.

5.6.5.5. Waterjet Repulsion

The evaluation of the waterjet repulsion concept quickly concludes that this concept is feasible for ships close together at low speeds, but is not feasible for a lateral separation distance of 20 feet or speeds of 8 knots and above. The shearing vectors the waterjets would encounter at speeds of 8 knots and above would render the waterjets ineffective. The energy and pumping system, together with the large flow volume required, make this concept impractical.

5.6.5.6. Static Mixer Repulsion

The evaluation of the static mixer repulsion concept also concludes that it is not feasible for the combination of speed and lateral distance required. Unlike the waterjet concept, which can be regulated and controlled by pressure and flow variations along with on/off control, the static mixer relies on the forward velocity of the host ship to provide the energy needed to operate.

6. Cargo Transfer Technology

This section focuses on concepts and technology related to the transfer of cargo in a skin-to-skin operation. Potential operational scenarios and types of cargo involved were examined in Section 3.0 of this report. Topics for this section include the feasibility and applicability of advanced pedestal crane systems, gantry cranes, rigid arm cranes, liquid cargo transfer, and the transfer of items via a trans-ship bridge/cargo shuttle/ILP.

6.1. Pedestal Crane Systems

6.1.1. Introduction

This section discusses how well various ship mounted pedestal crane systems would function during skin-to-skin operations. The large number of ships potentially involved in these operations differ in length, width, as well as have cranes which have different boom lengths and are mounted in different locations. Clearly a large amount of permutations is possible in such a study to investigate the effect of various ship-to-ship combinations and different crane designs and control strategies. To condense this study to a reasonable form, three increasingly severe ship motion cases are studied. This range of ship motion will illustrate how crane performance will deteriorate with larger ship motion and how much the crane speed and power requirements will increase.

Currently, most ship mounted crane system have a Rider-Block Tagline System (RBTS) installed. A baseline study was performed to estimate what workspace such an RBTS system would have for the different ship motion cases and how much payload motion would result. The RBTS system uses taglines to pull the payload in towards the crane cab, and help stabilize the payload. However, depending on the cable tension and the cable geometry, it is possible for the RBTS taglines to become slack. As this occurs, the rider-block and payload motion can become very erratic and large. In the RBTS baseline study the workspace is assumed to be the (x,y) payload placements where the RBTS taglines do not go slack. Note that the RBTS is a passive method designed to help reduce the probability of severe payload motion by attempting to keep the rider block statically stable and changing the natural frequencies of the payload suspension cables. When trying to land a cargo on a second vessel, the payload motion relative to the second vessel deck (referred to as tracking error) will be due to the payload swing relative to the primary crane ship, the primary crane ship motion itself, as well as the target ship motion.

A second crane design has the RBTS system removed and uses an active control scheme to command the three crane degrees of freedom (slew, luff, and hoist) to keep the payload inertially fixed. Sandia National Labs has developing such a Pendulation Control System (PCS) and is to demonstrate a version of it during the fall of 2002. In essence, the PCS system keeps the payload steady and swing-free while the primary crane ship is undergoing a general six degree-of-freedom motion. For the skin-to-skin cargo transfer mission, this results in the payload hanging steady of the target ship deck, while the tracking error will depend solely on how large the target ship deck motion is. Payload tracking error is the relative motion of the cargo load to the desired landing spot on the target ship

To implement a PCS system the crane joints have to move at high rates to cancel large ship motions. This study considers a modified PCS crane that has a self-leveling base. Here the crane tower base is mounted on a two degree-of-freedom platform, which is able to compensate for the primary crane ship roll and pitch motion. The study will show how much the required crane joint rates are reduced with this crane modification and compare the power consumption to a regular PCS crane. However, note that the payload tracking errors with the target ship will be identical in this idealized study between the standard PCS and the self-leveling PCS crane systems.

Lastly, a control modification is discussed that would allow the payload to actively track a position on the target ship deck. To implement this, advanced sensors would need to be implemented to track accurately both the primary and secondary vessel six degree-of-freedom motions. For the scope of this study, only the crane control requirements are discussed which are needed to track a steady-state target ship motion. Many research questions remain to be answered which depend on a more precise knowledge of how these crane ships will actually move in a skin-to-skin situation. However, the end result here is that the payload tracking errors due to the target ship motion could potentially be canceled with such a control strategy.

Note that this report only discusses baseline control requirements to implement various crane design and control techniques. The figures will illustrate the required crane motions to perfectly compensate for the various ship motions. Sensor and drive system imperfections are not discussed in this study. However, they will naturally degrade the system performance slightly and must be taken into account in any final crane design. The goal of this section is to illustrate what the crane speed requirements would be for various ship motions and provide estimates of how much payload tracking errors would ideally result with the various crane and control technique combinations. Further, this report will focus on the issue of horizontal payload tracking errors. Given the dynamics of having a payload act as a spherical pendulum attached to the boom tip, stabilizing and controlling this motion provides the most challenging task. For swing angles of 10 degrees or less, the vertical and horizontal payload motion essentially decouple. How well a vertical payload motion can be controlled only depends on how fast and precise the hoist drive system is. Controlling the horizontal motion of the boom tip is crucial in the cargo transfer problem since these ships tend to roll and pitch at a frequency which is very near the resonant frequency of the payload spherical pendulum. If this were not controlled a near-resonant input will cause the payload pendulation to quickly become excessive.

6.1.2. Ship Motion Description

In the previously discussed LAMP modeling, the ship motions of LMSR, MPF, TACS and DDG51 vessels experience roll and pitch motions in a range from as little as less than 1 degree to 5 degrees and higher. The simulations constrained the two vessels to have the same horizontal translational motion and the same heading. The only free degrees of freedom were the roll, pitch and heave. Of these, the roll and pitch appeared to be the dominant motion.

For the purpose of this crane requirements study, three different sets of increasingly severe ship motions are used. A variety of influences such as the sea state, swell amplitude and frequency, as well as the ship heading will determine how much two skin-to-skin vessels will move. Thus, in this study a certain motion is simply assumed without determining what sea conditions and ship heading caused them.

Table 6.1 - Parameters Used for Ship Motion Cases

	Case 1	Case 2	Case 3
Roll Amplitude	1.0 deg	3.0 deg	5.0 deg
Pitch Amplitude	0.5 deg	1.5 deg	2.5 deg
Roll Period	12 sec	12 sec	12 sec
Pitch Period	13 sec	13 sec	13 sec

Table 6.1 shows the roll and pitch sinusoidal amplitudes and periods that are used. The natural frequency of the various crane ships studied can lie between 10 and 15 seconds. Medium periods of 12 and 13 seconds were chosen. Also, the LAMP modeling showed that the roll motion typically was excited more than the pitch motion. For this study the pitch was thus simply set to half the value of the roll angle. Again, no estimations are made in this study as to how two ships will move for a given sea condition and ship heading. The crane performances are show for a mild, moderate and severe ship motion condition.

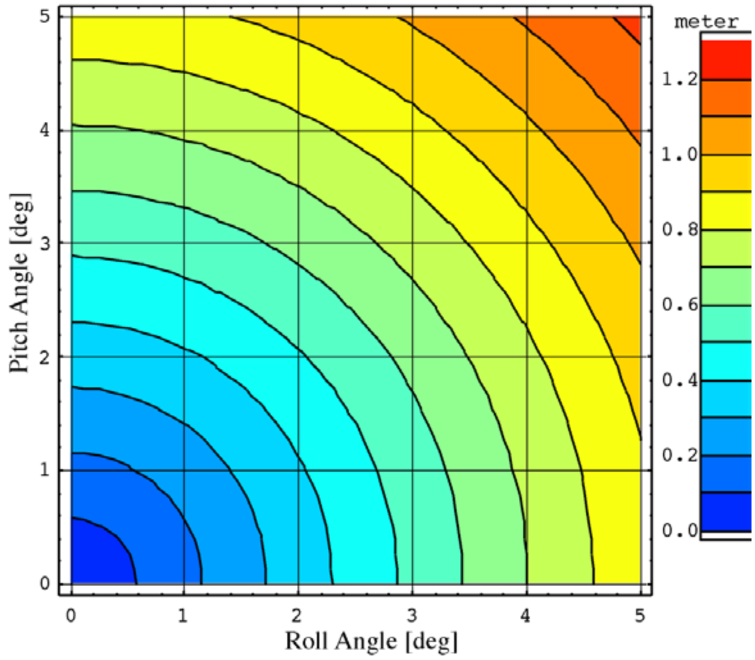


Figure 6.1 - Maximum Horizontal Translational Motion of a 10 Meter High Ship Deck Undergoing Various Roll and Pitch Rotations

The standard PCS and self-leveling PCS do not actively track the target ship deck. Instead, the payload is stabilized inertially and the moving target deck below it will determine the amount of horizontal payload tracking error. Figure 6.1 illustrates how much a deck will move horizontally if it is 10 meters above the pitch and roll rotation point. The roll and pitch periods of Table 6.1 were used to compute these maximum horizontal motion contours. The motion amplitude shown is the maximum radial distance from the steady-state target point. For example, if the target ship is rolling only 1 degree and pitching 4 degrees, then this figure shows that the horizontal deck motion would reach up to 0.7 meters radially, or about 1.4 meters from peak-to-peak.

6.1.3. Crane Types Considered

This section presents the various crane types that are discussed in this study. Note that the associated control strategies, crane speed requirements and payload tracking errors issues are discussed in the following section.

6.1.3.1. Rider-Block Tagline System

The Rider-Block Tagline System (RBTS) uses taglines to pull the payload hoist lines towards the crane cab and help reduce the payload pendulation that results from ship motion. If operating properly, the effective hoist length is the distance between the rider block and the payload. Since this pendulation frequency is much higher than the ship roll and pitch frequency, much less swing will result. However, note that even operating at its best this system will only keep the payload relatively fixed compared to the crane ship motion. As the ship rolls, for example, there may be less swing that is excited. However, the payload will still have some inertial motion, which can cause significant payload tracking errors. Thus, with the RBTS the primary crane ship motion is directly coupled to the payload tracking error.

Another issue is how well the RBTS system is able to keep the rider block in a ship fixed location. The RBTS has two out-riggers from which the taglines reach out to the rider block. The further apart these out-riggers reach, the more effective the taglines will be in stabilizing the rider block. However, the rider block can twist about the hoist line axis. If this twisting becomes too large, then slack taglines will result which typically causes large and erratic payload motion.

In this study the RBTS system is simply used as a baseline to compare the other crane control methods to. It is already installed on many shipboard crane systems and is thus the zero-cost upgrade option. However, it only passively attempts to keep payload swing to a minimum and not the actual payload motion (due to both swing and translation). Thus, using an RBTS system the payload tracking error will always be coupled to the primary and secondary ship motions. As this report will show, these errors can be rather large even without much swing being excited. Further, this report will provide estimates of the RBTS equipped crane workspace for various ship motion cases and payload masses.

6.1.3.2. Boom Crane with Pendulation Control System

The Pendulation Control System (PCS) crane has an active control system installed which is able to measure the six degree of freedom ship motion, as well as the two degree of freedom payload swing. Installing a PCS system requires upgrading the existing open-loop crane drive system to a closed-loop velocity servo system, installing a PCS control system, as well as installing the required ship motion and payload swing sensors. No major structural modifications need to be done to the crane. One six degree of freedom ship motion sensor is sufficient to cover the ship motion sensing requirements of all the cranes installed. However, a payload load swing sensor would need to be installed on each PCS equipped crane.

The goal of the PCS crane is to keep the payload inertially fixed. Being inertially fixed or referring to a coordinate frame as being inertial means that it is non-accelerating. A coordinate frame associated with the two skin-to-skin ships moving at a constant 8 or 16 knots, for example, would be an inertial frame. As the ship rolls and pitches, then the PCS will command the three crane degrees of freedom to compensate and keep the payload steady and swing-free as seen by the non-rotating, inertial frame. In essence, it decouples the crane ship motion from the payload motion. The end result is that despite the crane ship rotating at a near-resonant frequency of the payload spherical pendulum, the payload is held steady without introducing swing.

For skin-to-skin cargo transfer this means that the payload tracking errors will be drastically reduced. Ignoring the payload swing of an RBTS system and the issues of slack taglines, the RBTS attempts to keep the payload fixed relative to the moving crane ship. Assume that both the primary and target ship decks are moving horizontally up to 1 meter. Since these motions will not be in phase, the total tracking error can grow up to 2 meters here. If the crane ship motion is canceled by the PCS system, then only the 1 meter motion of the target ship remains. More realistically, the RBTS crane would have the 2 meter relative ship motion and any swing induced payload motion added to it. If the target ship is blocked from the incoming waves by the primary crane ship, then the roll and pitch of the target ship is often much smaller than that of the crane ship. Decoupling the crane ship motion from the payload tracking errors will then lead to an even higher percentage reduction in payload error motion relative to the target ship deck. Besides substantially reducing the tracking errors, the PCS system also makes the payload motion relative to the target deck smother and more predictable. If both ships are contributing to the payload motion relative to the target deck, then the motion will appear to the support crew as erratic and difficult to predict. Both ships are rolling and pitching in different manners, which cause the tracking errors to grow and shrink in an apparent non-sinusoidal manner. With the PCS system the crane ship motion is decoupled from the payload tracking errors. The tracking errors that a support crew aboard the target ship would perceive are now solely due to the motion of the target ship itself. Since at any instance the target ship's roll and pitch behavior is near sinusoidal, the tracking error will be perceived as being smoother and near sinusoidal. This will make it easier to land the payload and increase safety to the crew.

6.1.3.3. Self-Leveling Boom Crane with Pendulation Control System

The standard PCS crane must use its own slew, luff and hoist degrees of freedom to compensate for the crane ship motion. In extreme sections of the workspace the boom crane geometry may result in rather high crane joint rates for large ship motion cases. For example, consider the boom operating at a low boom angle. To compensate for a 1 meter ship motion, the boom will have to luff a large amount. At a zero boom angle the required luff rates will grow infinitely large. For large boom angles, the required luff motion for a given ship translation will be much smaller. Following sections will illustrate this point further in the required slew and luff rate contour plots. Also, the crane cab is rotating with the ship during these operations. If large ship roll and pitch motion is experienced, this could be awkward and discomforting for the operator.

To ease these issues, a PCS crane concept is investigated where the crane base is mounted on a self-leveling platform. This platform would have two degrees of freedom to compensate for the ship roll and pitch. Thus, the self-leveling PCS crane would still translate due to the roll and pitch of the ship, but it would remain level (relative to the gravity vector) while doing so. The end result is that the crane joints will not need to be as fast as on a regular pedestal crane. This would ease structural issues of moving the crane at the higher PCS rates. Also, the operator is now sitting in a cab that is always level, despite how much the ship is rolling and pitching.

To implement such a system all the previous PCS requirements would need to be met. Further, the crane base structure would have to be rebuilt to accommodate the moving base. Additional hydraulic systems would need to be installed to provide the large torques required to keep the crane level. Keep in mind that the boom and the payload are applying a large combined torque onto the moving base. The hydraulic system would have to provide a constant counter torque to keep this platform level. As the ship moves, the hydraulic system would then have to move the base accordingly to compensate for this motion. A later section will illustrate how the crane joint rates would be affected and how the total power usage compares to the standard PCS implementation.

Note that the goal of implementing the self-leveling PCS crane is to reduce the crane joint rates, and thus increase the crane workspace for a given limit on these crane joint speeds. However, if operating with ideal sensors and crane actuators, both cranes will result in the same amount of payload tracking error. Both cranes decouple the crane ship motion from the payload error motion. Only the target ship motion would contribute to the tracking errors. The potential increase in workspace is achieved at the expense of designing, building and implementing an entirely new crane base structure.

6.1.3.4. Rigid-Arm Crane System

As mentioned earlier, having the payload suspended from the boom tip as a spherical pendulum makes payload motion control more difficult. A natural question to arise is, couldn't the boom crane with the hoist cable be replaced with a rigid set of links that directly control the payload position? Such robotic manipulator systems are routinely used in the industry to assemble vehicles and perform other tasks. However, due to the payloads involved and the large reach required, serious structural issues arise to implement such a system on a ship. With any manipulator (boom crane or robotic arm), each link must be able to carry its own weight and the weight of all the links attached to it, as well as the end payload. With the boom crane, the link from the boom tip to the payload is a hoist cable system. Cables are very light weight structures whose mass can essentially be ignored compared to the remaining crane structure. Thus, this light weight structure is able to support the necessary lifting requirements to carry a large payload without contributing significantly to the mass and strength requirement of the link that it is attached to (boom in this case). For the TG3637 crane found on the T-ACS, a 24000 kg boom is able to support a 36000 kg payload. However, the drawback is that the cable link is not rigid and is free to swing. A light weight structure is obtained at the expense of less rigidity. Also, the boom structure is assumed to carry the payload weight in compression. This is achieved through a specific geometry of the luff and hoist cable rigging. Thus, it can only carry this payload along the gravity vector. The crane structure is not strong enough to be able to support the payload hanging from the boom tip with large swing angles.

If a set of rigid links were to replace this boom-hoist cable system, it would have to be strong enough to carry the large payload at a variety of angles. A minimum of two links would be required to place the payload at arbitrary points (slew mode is ignored in this simplified example). Since the current crane boom, weighing 24000 kg, can only carry the payload weight in compression, the link placing the payload would have to be significantly larger than this. The primary link, which would support the weight of the end link and the payload, would have to be even stronger and more massive. The ratio of the required manipulator mass to payload mass quickly grows very large as the payload mass and required operation range increase. Rigid arm cranes' available payloads are under 10 LT, making them unsuitable for the task of lifting containers and other heavy cargo. A cable based manipulator system, such as the boom crane design, appears to be a more feasibly solution to achieve the required lift capacity and operating range. Thus, for this section, the control requirements for using potential rigid-link cranes were not considered. A discussion of rigid-link crane capabilities will follow.

6.1.4. Control Strategies

This section outlines various control strategies and illustrated the required crane rates and total power consumption. Also, workspace plots are provided to illustrate where the crane would be able to operate for a given set of crane servo speed limits.

6.1.4.1. Rider-Block Tagline System

The RBTS system does not contain an active control system, nor does it require any expensive sensor systems. Instead, the rider block is set at a fixed location, which is determined through the in-haul angle and tagline angle. These two angles, as well as the operation range of the rider block with a heavy payload, are illustrated in Figure 6.2. The tagline angle was set to zero degrees for all cases considered. This represented a reasonable value between the minimum (-15 degrees) and maximum (30 degrees) tagline angles described in the operator manual.

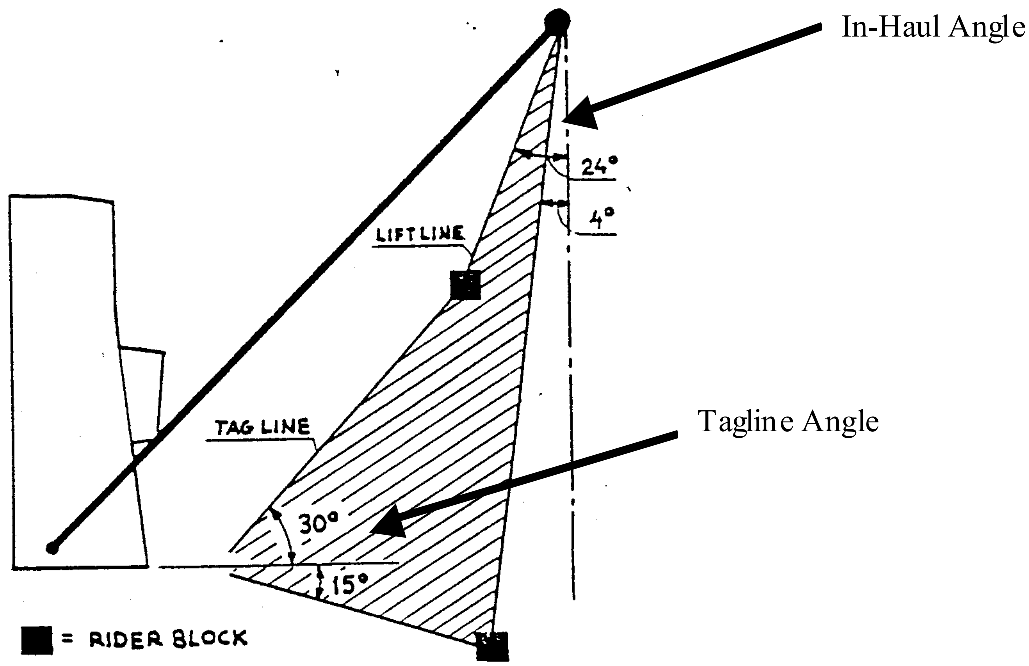


Figure 6.2 - Illustration of the RBTS In-Haul and Tagline Angles

According to the TG3637 RBTS operating manual, the allowable in-haul angles for a large payload are set using Figure 6.3. This fixes the rider block location in the most rigid manner. During the simulations, lines (1) through (3) in Figure 3 were used to determine the appropriate in-haul angle if a large payload is suspended. If a small payload is suspended, then line (2) is increased to yield an in-haul angle of 36 degrees with line (1) being stretched accordingly. This avoided large RBTS-hook block double pendulation issues when the smaller inhaul angles were used.

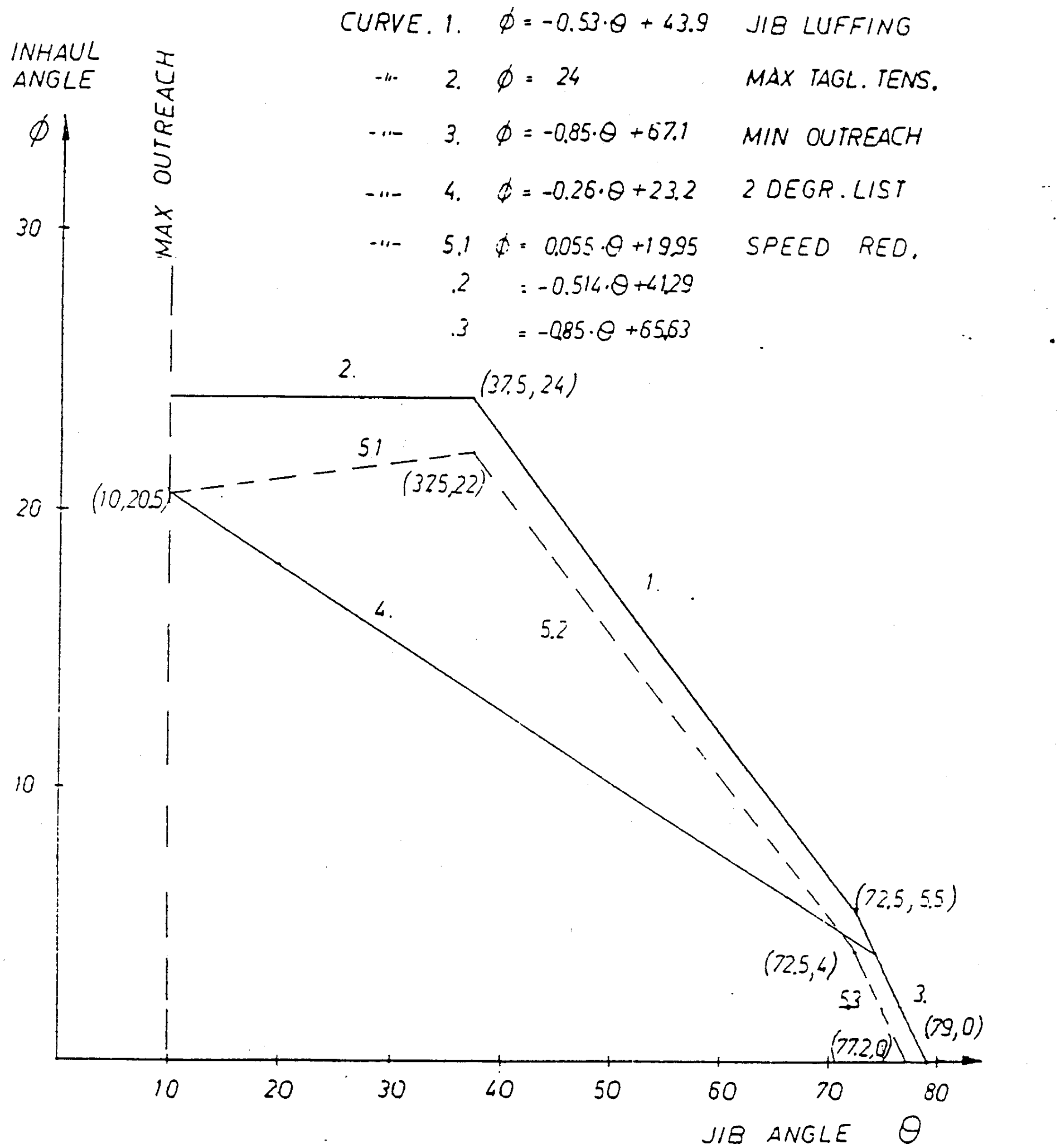
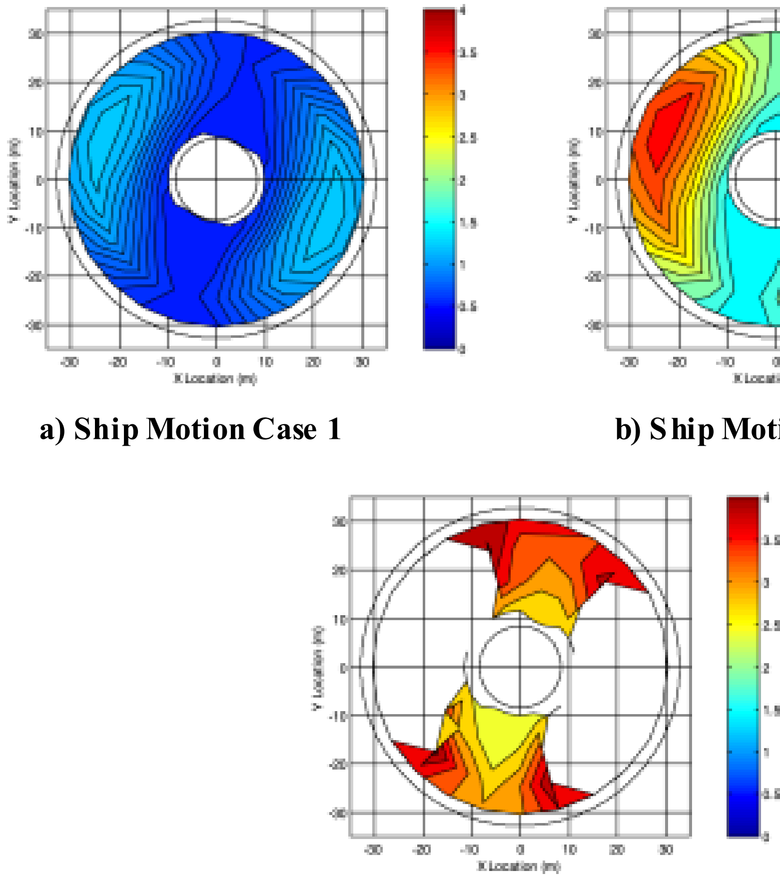


Figure 6.3 - Illustration of the Jib Angle and Allowable In-Haul Angles.

Using a TACS-5 vessel with the TG3637 crane and installed RBTS, 6 sets of numerical simulations were performed. Each simulation sweeps the payload through the potential workspace of the crane and determines the amount of inertial payload motion that would result, as well as determines if slack taglines would occur. According to the operating manual, slack lines are defined as the cable tension being less than 2500 N. If this occurs, then the simulation sets this location outside of the available workspace. The three ship motion cases in Table 6.1 are used for a large payload of 20 long tons and a small payload of 7500 kg. The small payload corresponds to just trying to place the hook block with a spreader bar attached. The payload is always assumed to be at deck level, which was set to be 10 meters above the water line for all vessels. Note that no interference issues with the taglines were modeled here. If the task is to reach across a large Panamax class container ship, it might be impractical to pick up a far container first because the taglines would snag on the near containers. To reach the far container, it would then be required to first remove all near containers, which could slow down the cargo transfer process significantly and prevent selective off-loading.



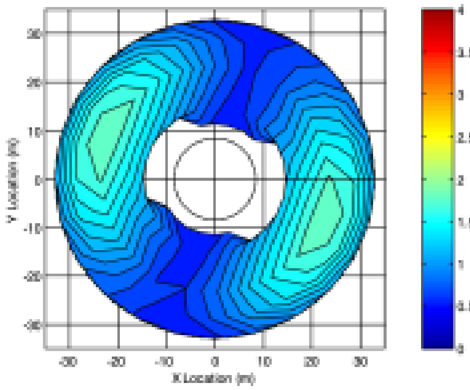
a) Ship Motion Case 1

b) Ship Motion Case 2

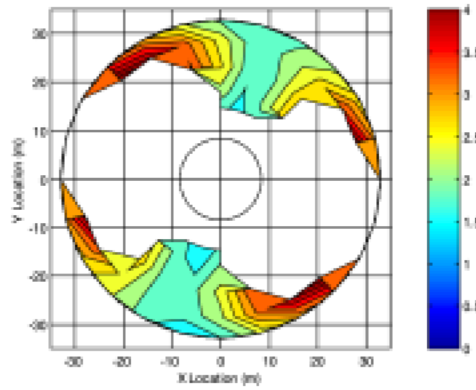
c) Ship Motion Case 3

Figure 6.3 - RBTS Workspace and Inertial Payload Motion Illustration for Ship Motion Cases 1-3 and a Small 7500 kg Payload (x-axis points towards bow, y-axis points toward port side)

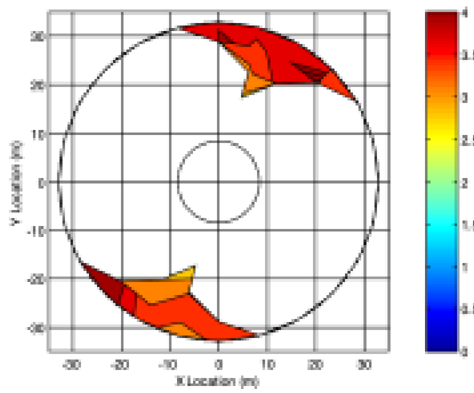
Figures 6.3 and 6.4 illustrate the simulated workspace of the TG3637 RBTS system for the different ship motion cases 1-3. The center of the coordinate system shown is the center of the crane. For each ship motion case, the 8 degree-off-freedom, rider block/payload simulation code was used to generate inertial payload motion using different ship motion profiles. Performance was quantified by recording the maximum horizontal payload motion, relative to its zero swing location, achieved during a 40 second. Note that this maximum payload motion is relative to the non-rotating ship frame. Sine the vessels are assumed to be moving at a steady 8 or 16 knots, this frame is inertial. To estimate how large the payload tracking error would be relative to the target ship, the horizontal target ship motion in Figure 1 would need to be added for a particular target ship roll and pitch amplitudes. The simulated crane is located (33.77, -8.3, 18.87) meters off the ship of the ship center. Note that each T-ACS simulation is run with the same crane location.



a) Ship Motion Case 1



b) Ship Motion Case 2



c) Ship Motion Case 3

Figure 6.4 - RBTS Workspace and Inertial Payload Motion Illustration for Ship Motion Cases 1-3 and a Large 20 long ton Payload (x-axis points towards bow, y-axis points toward port side)

Note that the light payload case results in similar amounts payload motion as the large payload case, while the available workspace differs noticeably. As the ship motions increase, areas of predicted slack taglines increase dramatically. However, the higher in-haul angles of the light payload case helps the RBTS performance by avoiding the early onset of slack taglines. With a heavier payload attached there will be very little double pendulation occurring. However, the heavy payload will cause the rider bock to twist. As the payload motion becomes sufficiently large, then slack taglines may occur here too. The simulations predict very significant horizontal payload motion for all three ship motion cases. The in-haul angles can not be increased as much with the large payload cases to avoid putting excessive stress on the RBTS out-rigger bars. In practice the payload might have to be stabilized with guide ropes manned by a support crew.

6.1.4.2. Pendulation Control System

The standard PCS crane uses ship motion sensors and swing sensors to stabilize the payload in inertial space. Effectively, the crane ship motion is being decoupled from the payload motion. Several sets of simulations were performed to illustrate how the crane speed and power requirements change for different ship motion cases. Note that these simulations assume a perfect drive system and no sensor errors are modeled. Thus, the following results present a baseline estimate of how fast and powerful a crane would have to be, as a minimum, to cover a given workspace. The actual PCS crane operates in closed-loop mode to reject any swing errors due to lift off transients or other disturbances. Depending on how good the crane drive system and sensors are, an additional 10-20% of crane speed would be required to implement a closed-loop control system.

Each page shows the results for a given ship motion case and crane case. A T-ACS vessel (23.2 meter beam) with a TG3637 crane (37.45 meter boom length) is considered, as well as a LMSR ship (31.2 meter beam) with a crane of boom length 40.2 meters. Also, note that the TACS cranes are mounted off the ship center-line by –8.3 meters, while the LMSR cranes are mounted along the ship center-line. This illustrates crane joint rate and workspace differences between having a crane mounted in the center of the ship and off to the side. To make the comparison simpler, both cranes are assumed to be 33.77 meters forward of the ship center and 18.87 meters above the water line. Doing so both cranes will experience similar roll and pitch induced translations and the effect of moving the crane to the center of the ship is more apparent.

Each figure page shows contour plots illustrating the maximum required slew and luff rates. Note that these are boom luff rates and not luff servo rates. The slew angles are varied here between +/- 180 degrees. The slew joint limits of +/- 90 are ignored in this study since the crane pedestal allows for any crane orientation to be achieved. The luff angles are only allowed to vary between 10 and 80 degrees. This corresponds to the luff joint limits present on the TG3637 crane. If a crane joint motion for a particular payload position requires luff angles outside of this range, then no joint rates are shown since this payload position is now considered outside of the available workspace.

The contour plots have two black circles. These indicate the absolute minimum and maximum outreach of each crane at the extreme luff angles. The ship is represented by a gray-line diagram. The LMSR crane is located in the middle of the ship, while the T-ACS crane is shown as being to the side.

The following two contour plots on each figure page illustrate the estimated power requirement to operate all crane degrees of freedom at the same time. A heavy payload case of 20 long tons and a small payload case of 5000 kg are shown. For the different crane control concepts these power requirements can vary drastically with payload. With the given choice of payloads either extreme is covered. The power calculations are simple estimates where the static load on the hoist and luff servo drums are assumed to be constant. The variations in luff and hoist servo power requirements due to the PCS have been bound to be small compared to the static load induced power requirements of these modes.

The final figure illustrates the estimated crane workspace for the given ship motion case. Here it is assumed that the slew rate limit is 6.5 degrees/second and the luff winch drum speed limit is 342 degrees/second. Depending on the boom luff angle, this luff servo rate corresponds to a maximum luff rate of 3-4.5 degrees/second. These values are picked from the proposed enhanced TG3637 crane drive system that is to be installed on the T-ACS 5, SS Flickertail State. Note that these values are simply chosen as an example. It would be possible to add additional pumps to increase the available joint speeds further. However, by choosing to keep the same joint speed limits across various crane and control designs, the differential effect of adding a self-leveling crane base, for example, is more apparent. If the ship motion will cause the crane to hit a joint limits, these workspace boundaries are illustrated as a red contour line. Luff servo rate limits or slew rate limits are shown as green and blue contour lines respectively.

The T-ACS ship motion case 1 plots show that the PCS crane would be able to cover the entire workspace with the chosen slew and luff rate limits. Note that the slew rates will always be at their largest value at 0 or 180 degree slew angle with large luff angles. In contrast, the luff rates are at their largest at +/- 90 degree slew angles and small luff angles.

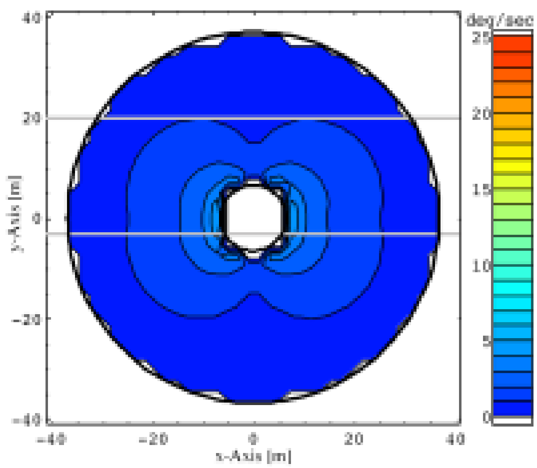
As was mentioned earlier, this study assumes perfect sensors and crane drive systems. Thus, this control would be able to compensate perfectly for the crane ship motion. Payload tracking errors relative to the target ship are solely due to the target ship motion. Studying Figure 6.1, we see that 1 degree roll and 0.5 degree pitch would result in up to 0.2 meters of horizontal motion of a deck with height of 10 meters. We are assuming here that the target ship is experiencing motion of the same order of magnitude as the primary crane ship. For this ship motion case the conclusion is that this idealized PCS control strategy could reduce the payload tracking error to the 0.2 meter level.

Actual system performance would degrade due to sensor and crane drive system limitations. Compare this result to the tracking errors of the RBTS study. For ship motion case 1, the inertial payload motion is seen to reach up to 1-1.5 meters. Including the same target ship motion, this value increases to 1.2 – 1.7 meters.

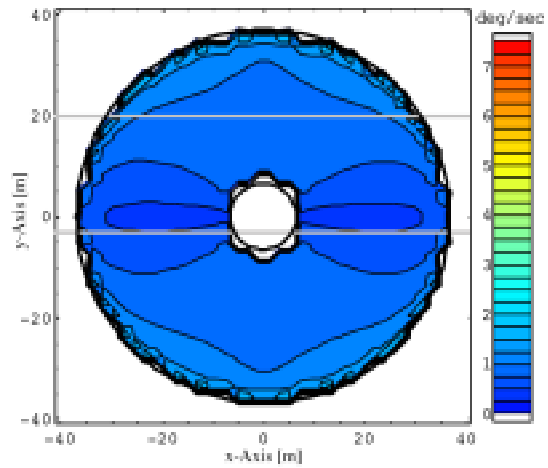
The total power consumption of the PCS crane shows a very interesting behavior. With the heavy payload of 20 long tons, there are few contour lines visible across the workspace. This means that the total power consumption is near constant across the workspace. If one were to plot the individual power requirements for the luff and hoist modes, then one would observe these to increase to 200-300 kW levels. However, since the pumps for each crane degree of freedom are driven off the same planetary gear, negative power requirements are fed back to the system through this planetary gear system. A loss rate of 20% was assumed in feeding back negative power to offset the positive power requirements. For example, if the boom is being lowered a negative power is required. With the PCS crane the hoist mode typically requires a positive power rate as the luff is providing a negative power rate, and vice versa. This results in essentially a power balance over the crane workspace as shown. To visualize this, consider moving two elevator carts which are connected through a pulley system. Raising only one cart by itself would require a lot of power, power being essentially the joint speed times the joint load. However, if both carts are connected through the pulley system, then it will take far less power to move the same cart since the second cart will act as a counter balance. The PCS control strategy attempts to keep the payload inertially fixed. This results in the crane moving in specific ways such that the power consumption demands balance each other. In contrast, if a smaller payload is attached then the system is no longer balanced. Lowering the payload (negative hoist power) does not provide sufficient power to the luff mode, which will need to be typically raised at this point. In this case we see the power requirements increase radially away from the crane center.

As the ship motion increases in severity for the higher ship motion cases, the rate and power requirements will increase as well. For the given joint limits, the available workspace is clearly reduced with the ship motion case 2. With ship motion case 3 the given joint limits result in a very small workspace. To increase the workspace, the slew or luff joint rate limits would need to be increased. The joint rate contour plots illustrate how fast the crane has to move to reach certain workspace areas. Workspace areas bounded by a red line can never be reached due to the crane hitting the luff joint limits here.

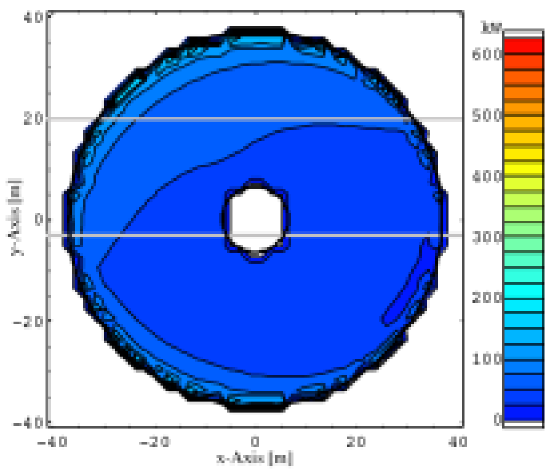
Moving the crane to the center of the vessel, as is the case with the LMSR case study, results typically in a more symmetric rate and power distribution across the ship roll axis. However, the rate and power requirements do not change substantially. The T-ACS crane is limited in its ability to reach far across the ship on the port side, but it has an easier time reaching far on the starboard side. The benefit is that it will have an easier time reaching container on the far side of a large Panamax class container ship (30 meter beam) if it is parked on the starboard side. The LMSR type of crane will be able to service vessels equally well on both sides, but it does not gain the extra reach that the side mounted T-ACS crane enjoys.



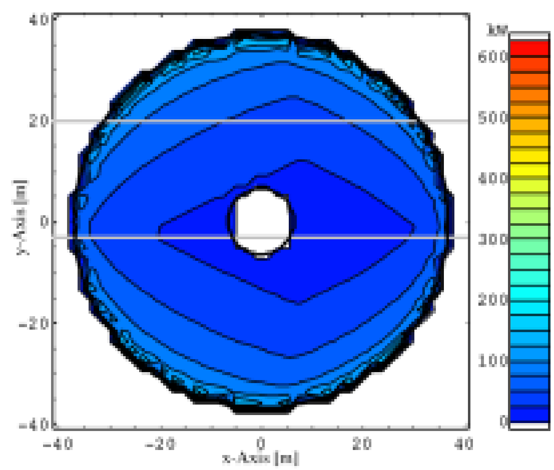
a) Maximum Slew Rates



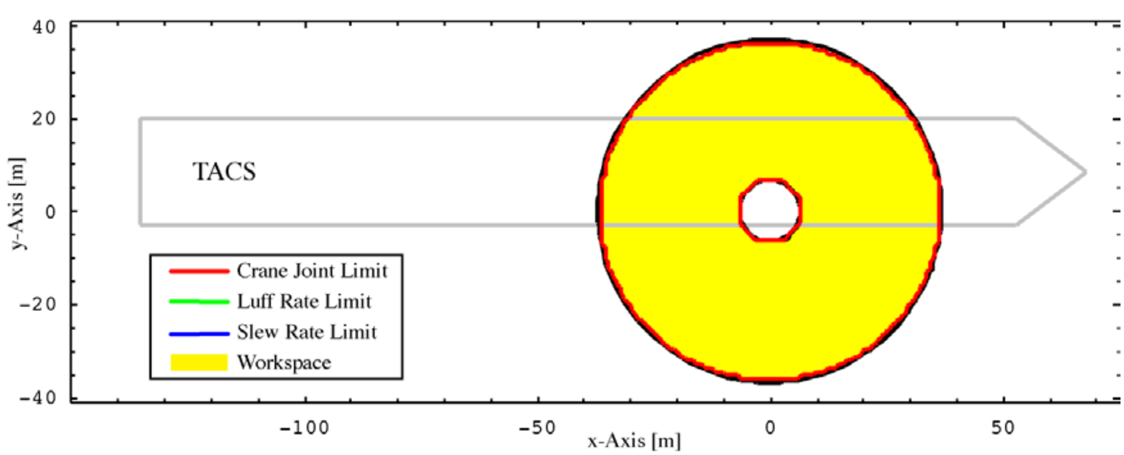
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

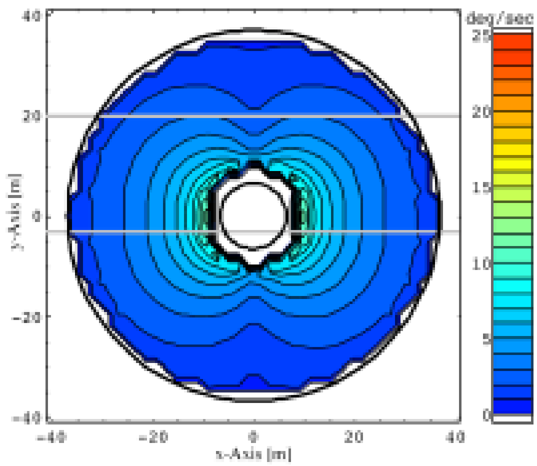


d) Maximum Total Power with 5000 kg payload

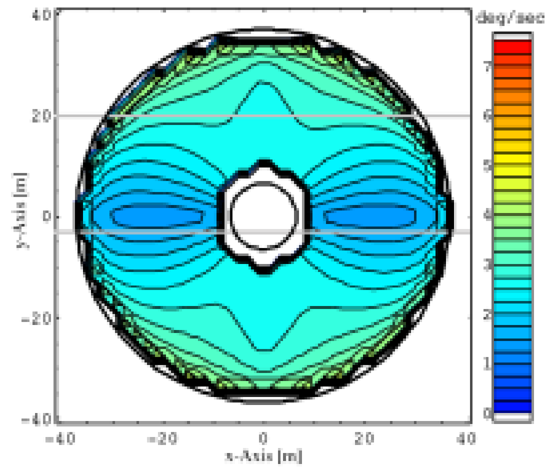


e) Workspace Plot Showing Joint and Crane Rate Limitations

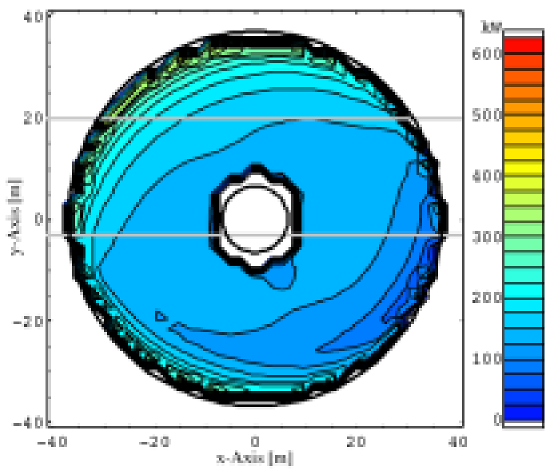
Figure 6.5 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a T-ACS PCS Crane



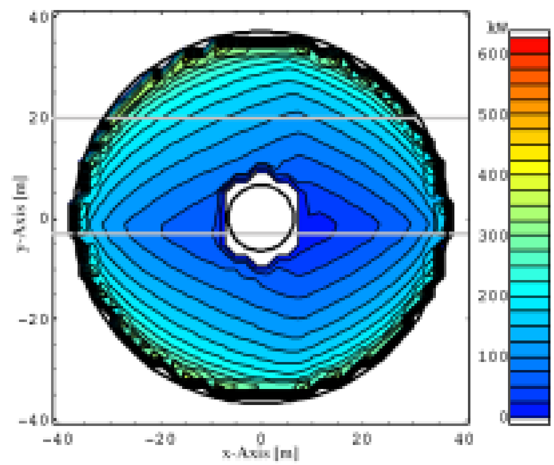
a) Maximum Slew Rates



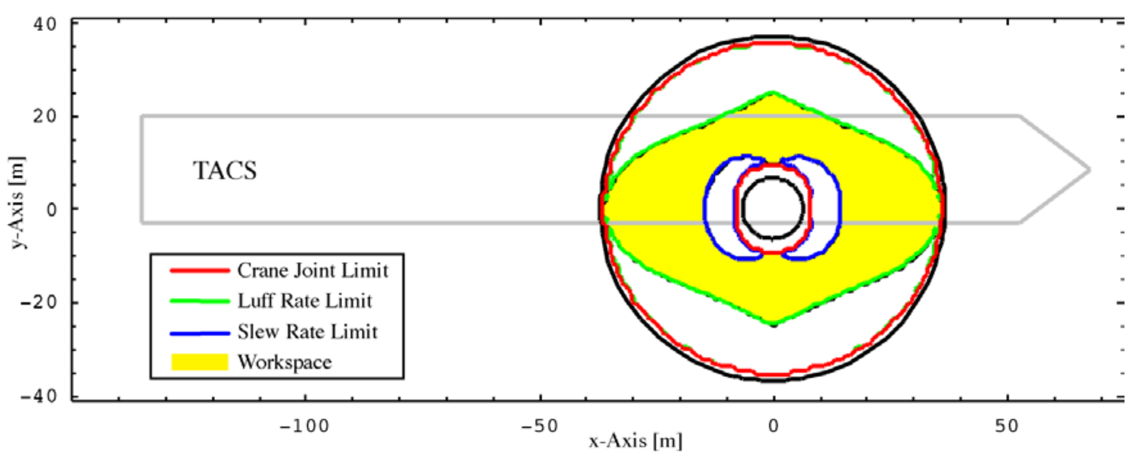
b) Maximum Luff Rates



c) Maximum Total Power
with 20320 kg payload

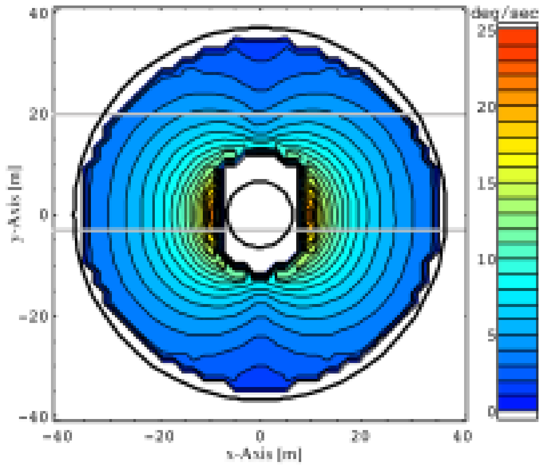


d) Maximum Total Power
with 5000 kg payload

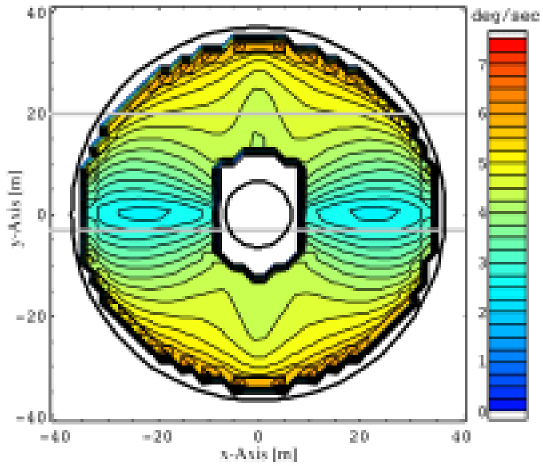


e) Workspace Plot Showing Joint and Crane Rate Limitations

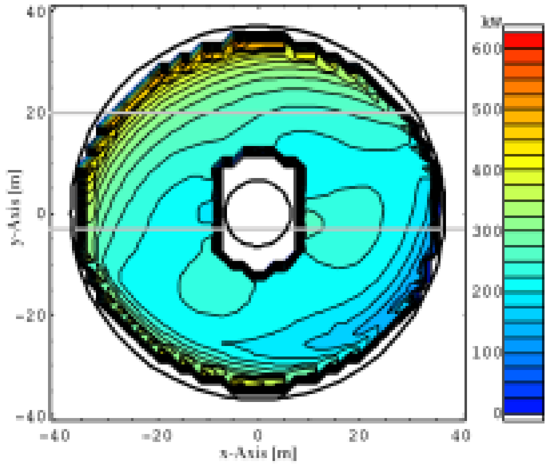
Figure 6.6 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a T-ACS PCS Crane



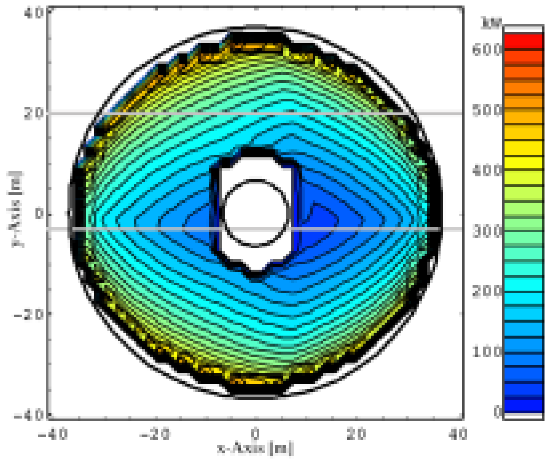
a) Maximum Slew Rates



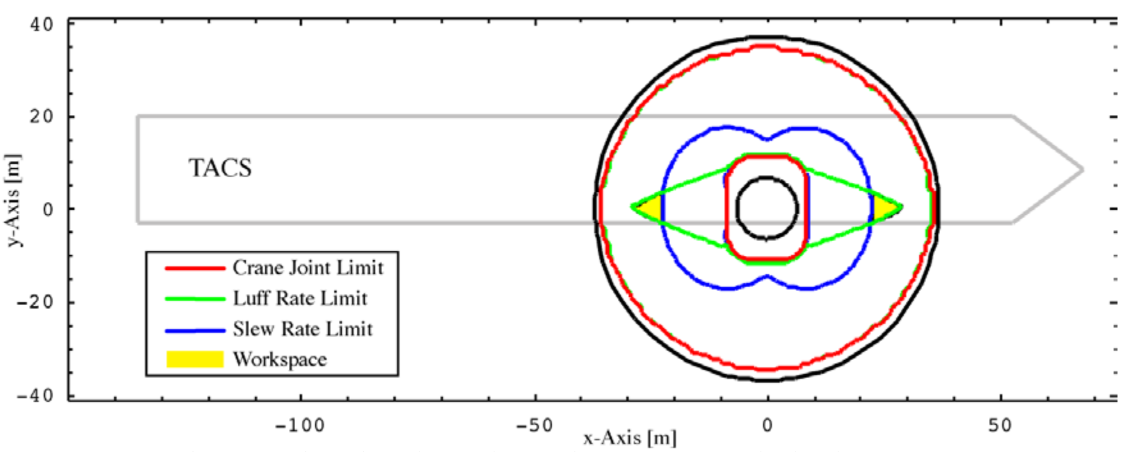
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

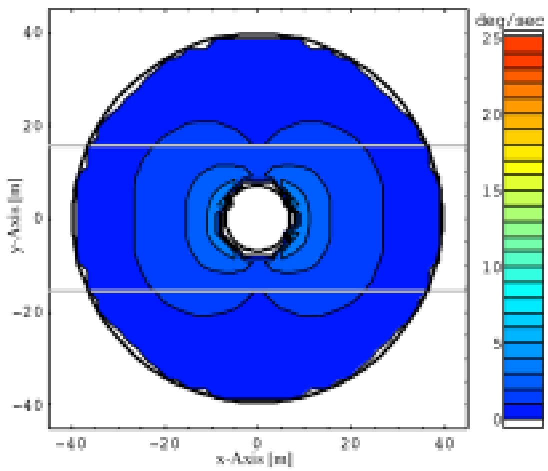


d) Maximum Total Power with 5000 kg payload

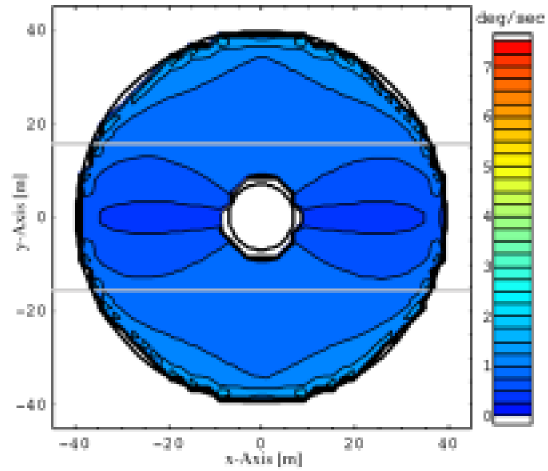


e) Workspace Plot Showing Joint and Crane Rate Limitations

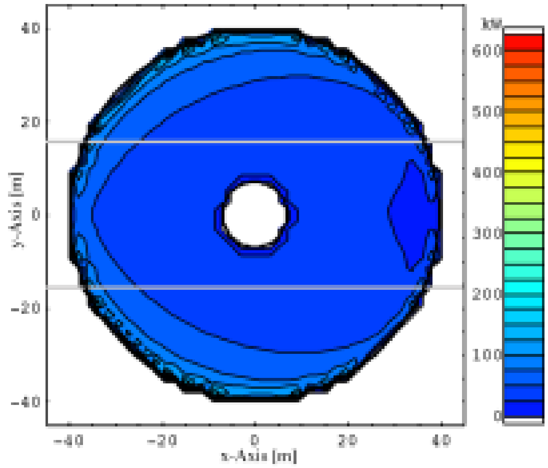
Figure 6.7 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a T-ACS PCS Crane



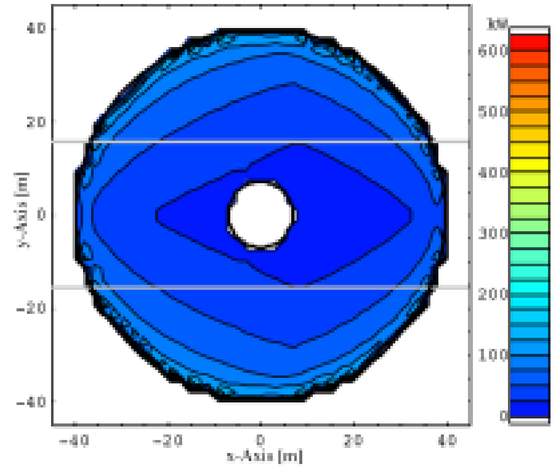
a) Maximum Slew Rates



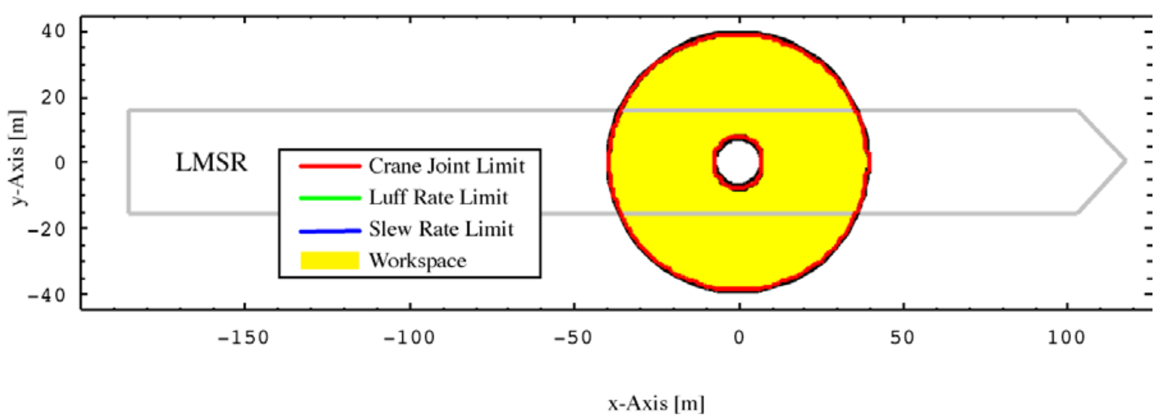
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

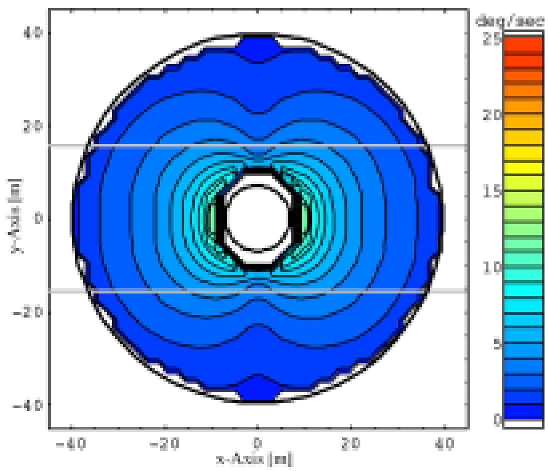


d) Maximum Total Power with 5000 kg payload

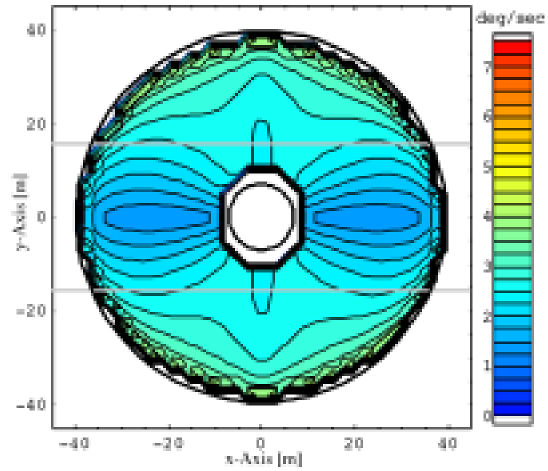


e) Workspace Plot Showing Joint and Crane Rate Limitations

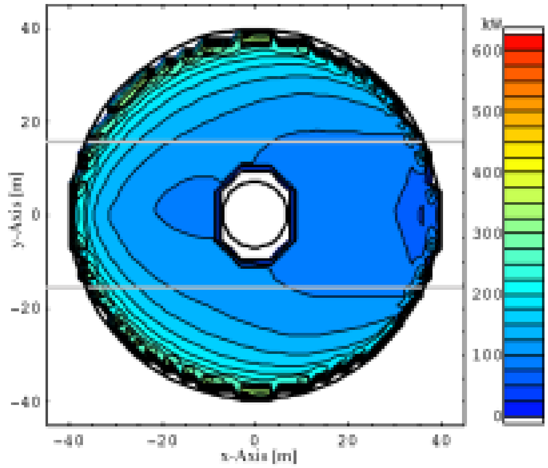
Figure 6.8 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a LMSR PCS Crane



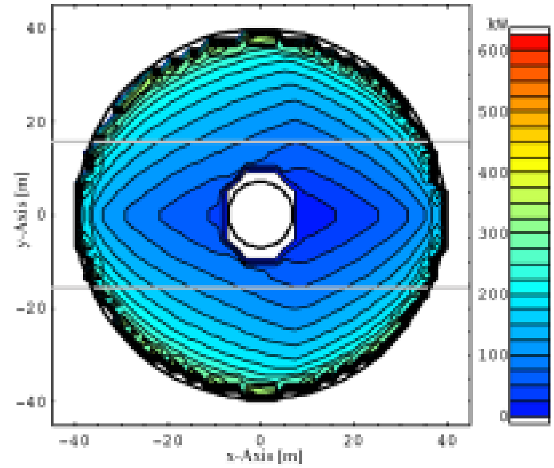
a) Maximum Slew Rates



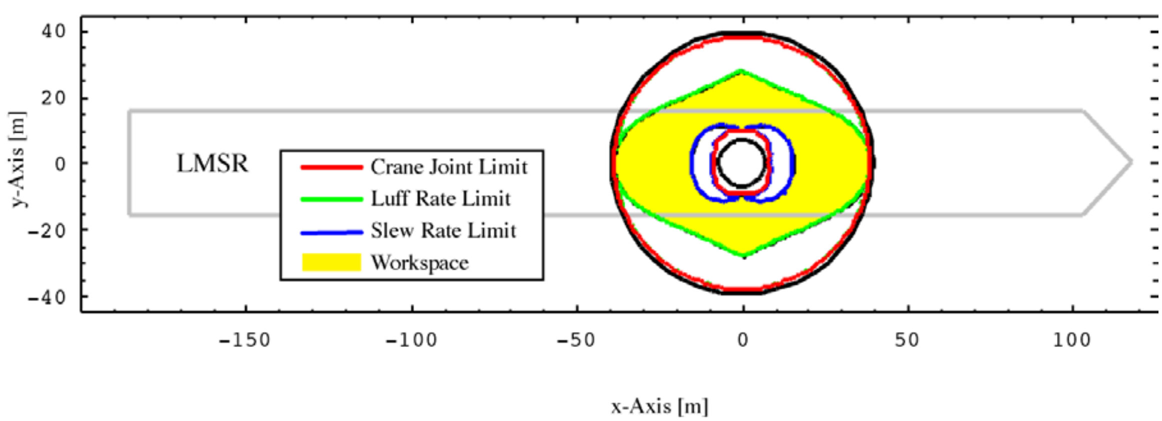
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

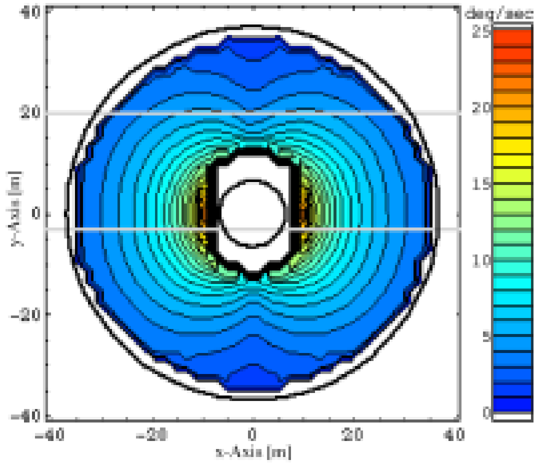


d) Maximum Total Power with 5000 kg payload

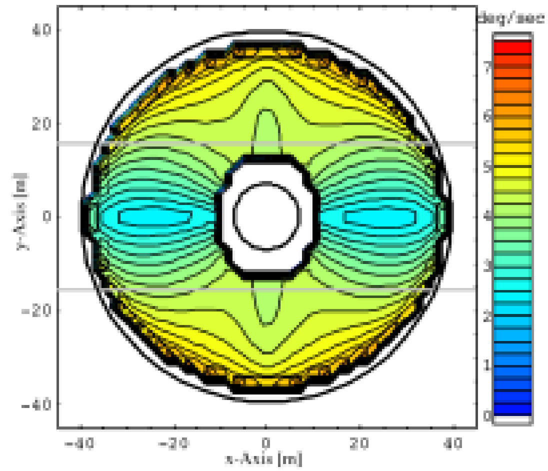


e) Workspace Plot Showing Joint and Crane Rate Limitations

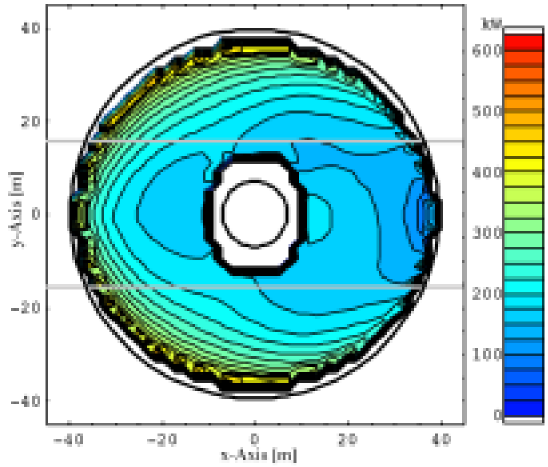
Figure 6.9 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a LMSR PCS Crane



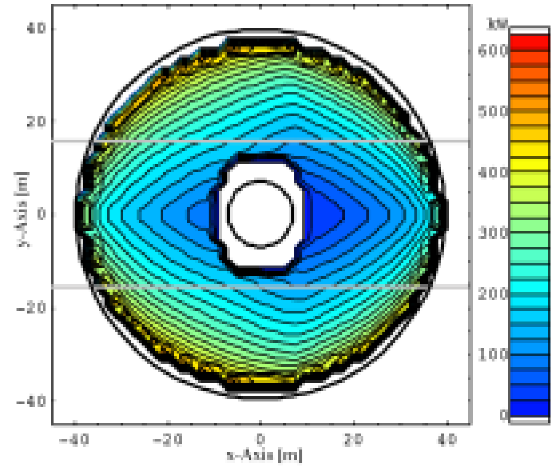
a) Maximum Slew Rates



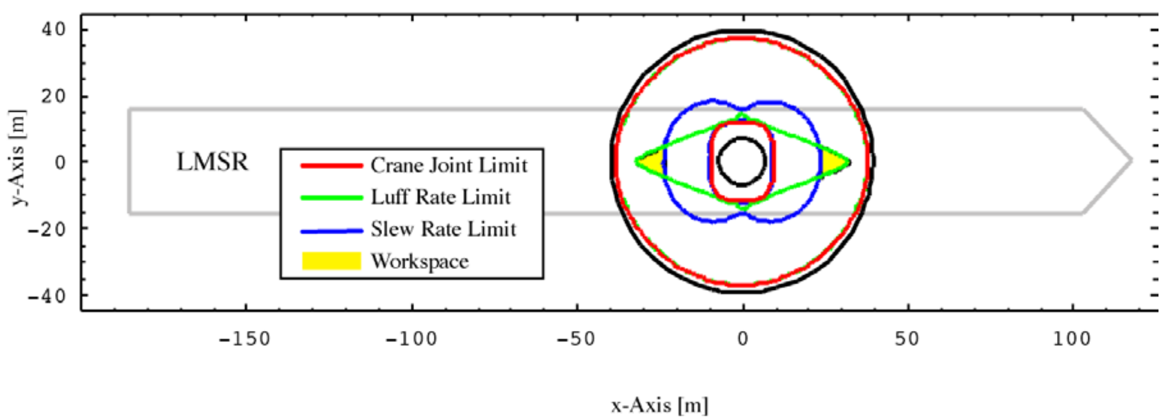
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload



d) Maximum Total Power with 5000 kg payload



e) Workspace Plot Showing Joint and Crane Rate Limitations

Figure 6.10 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a LMSR PCS Crane

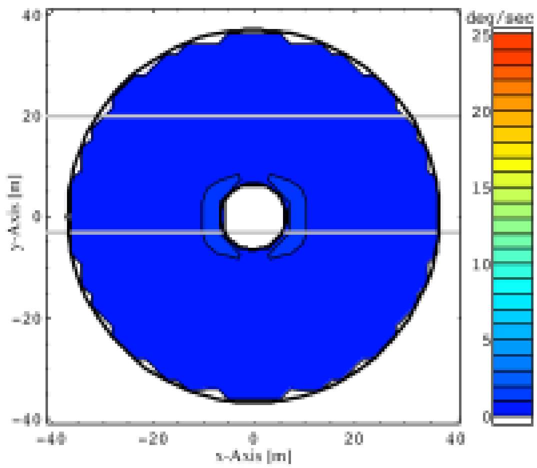
In conclusion, standard pedestal mounted boom cranes with the Pendulation Control System installed are an excellent option for skin-to-skin cargo transfer. Within the range of motions expected, PCS cranes will be capable of operating safely and efficiently. An advantage of the PCS system is that the cranes do not require significant modifications. Many currently installed cranes will likely require machinery upgrades to meet the required performance levels, but structural changes to the crane or the ship would most likely not be necessary

6.1.4.3. Self-Leveling Crane with Pendulation Control System

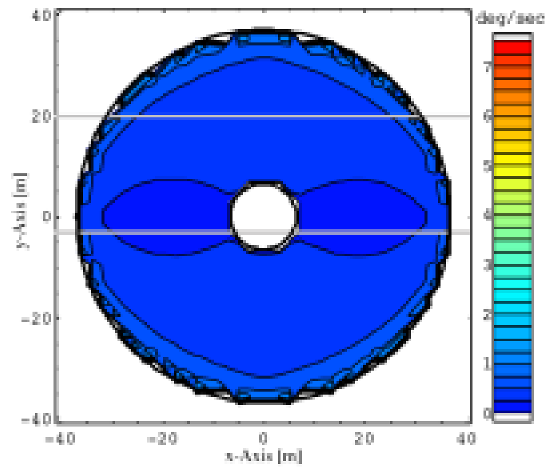
The simulation runs for the PCS crane are repeated here where the crane base is assumed to be self-leveling. To compute the total power requirements, the static torque that the boom and payload will apply on the crane house was computed and multiplied by the required platform roll and pitch rate and added to the crane joint power requirement. Since the base is assisting in compensating for the ship roll and pitch motion, the crane slew, luff and hoist degrees of freedom must now only compensate for the roll and pitch induced translational motion. As a result, all the crane joint rate requirements have been reduced substantially for both the T-ACS and LMSR crane case. Due to these lower slew and luff rate requirements, the workspace for different ship motion cases has been increased substantially.

The total power requirement for this self-leveling PCS crane is very different to that of the standard PCS crane. The standard PCS mode of operating has the power balance between the luff and hoist mode keeping the total power rates from growing too large. The self-leveling platform must continually apply a large torque to offset the large moment arm of the boom and the payload loads. As the platform is rotating, this will result in a larger power requirement that is not offset by a negative power requirement of another crane degree of freedom. The result is that the power requirements of the self-leveling PCS crane will typically be noticeably higher than the standard PCS crane power requirement for the large payload case. However, for a small payload the payload moment effect is substantially reduced and so is also the total power requirement. For ship motion case 1, the estimated power requirement for the self-leveling PCS crane is lower with a smaller payload than that of the standard PCS crane. To compensate for the higher power requirements while lifting a heavier payload, this self-leveling crane design could be incorporated with a counter weight to offset the large payload induced moment on the crane tower base. Employing hydraulic accumulators could also be used to further reduce the power consumption of the base.

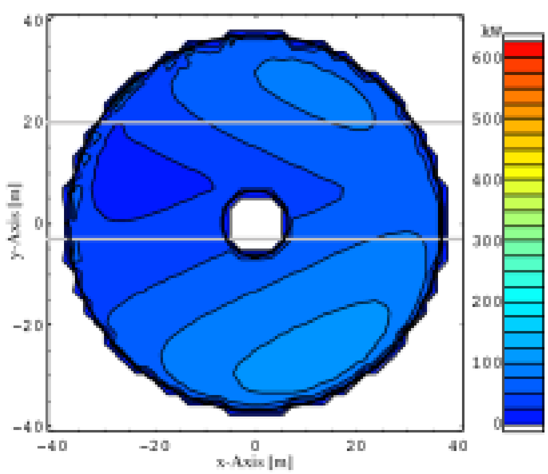
Installing a self-leveling crane base would provide an engineering challenge. The support structure and crane base might have to be redesigned to be able to handle the new loads. However the benefits in performance could be significant. The crane would not have to move as fast, which would lessen boom structural concerns. Further, the crane operator is also sitting on a level platform instead of a rotating platform. Most importantly, the workspace is drastically increased for a given set of crane joint limits. The larger the ship motions are, the more noticeable the difference in workspace becomes. Also, recall that the self-leveling PCS crane does not provide better payload tracking errors than the standard PCS crane. Only the required crane rates and associated available workspace are different.



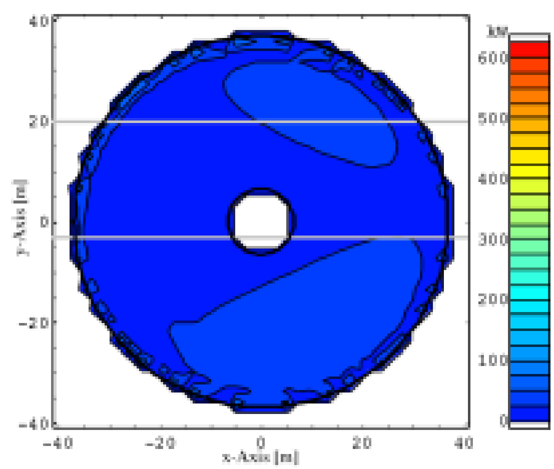
a) Maximum Slew Rates



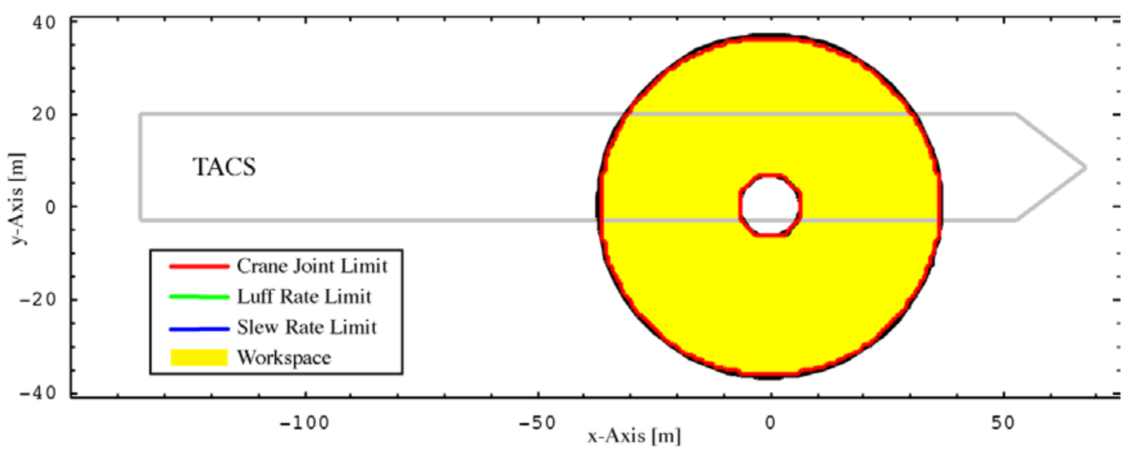
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

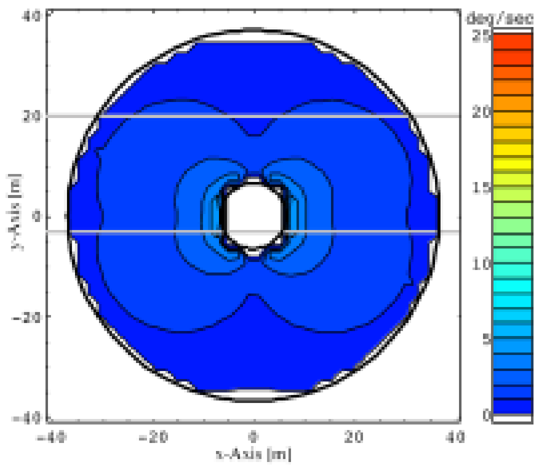


d) Maximum Total Power with 5000 kg payload

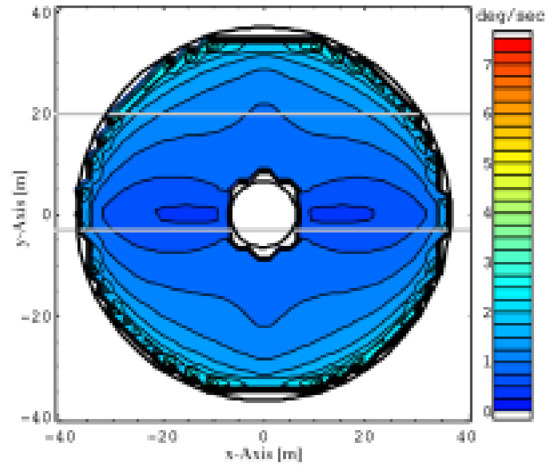


e) Workspace Plot Showing Joint and Crane Rate Limitations

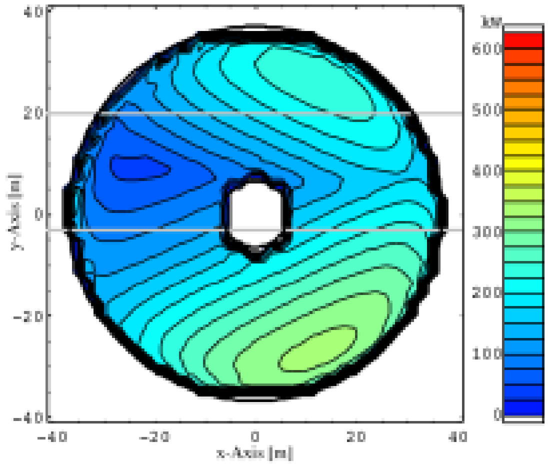
Figure 6.11 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a TACS Self-Leveling PCS Crane



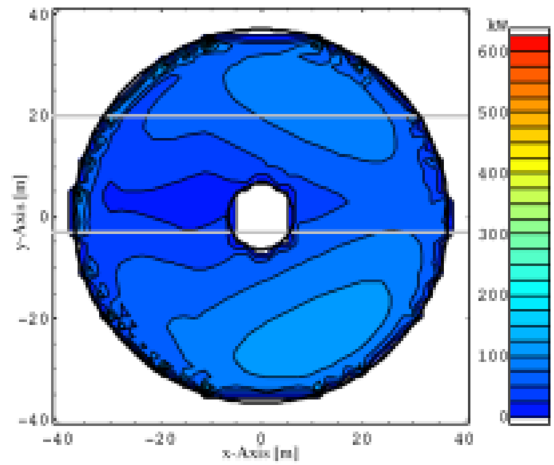
a) Maximum Slew Rates



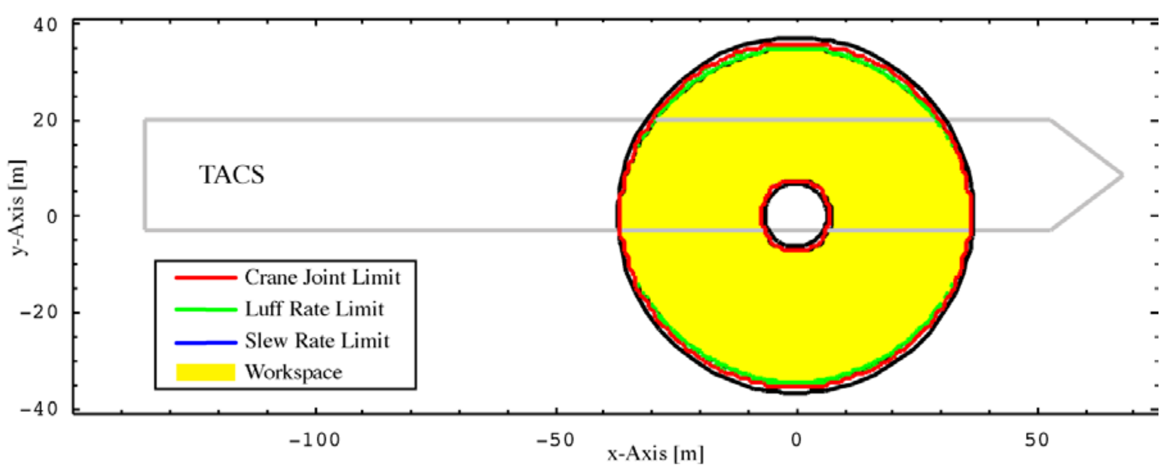
b) Maximum Luff Rates



c) Maximum Total Power
with 20320 kg payload

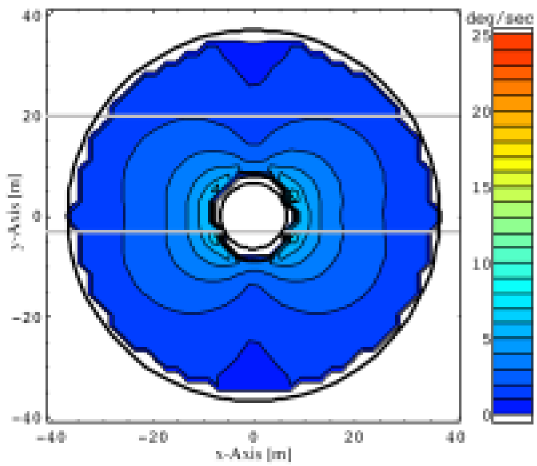


d) Maximum Total Power
with 5000 kg payload

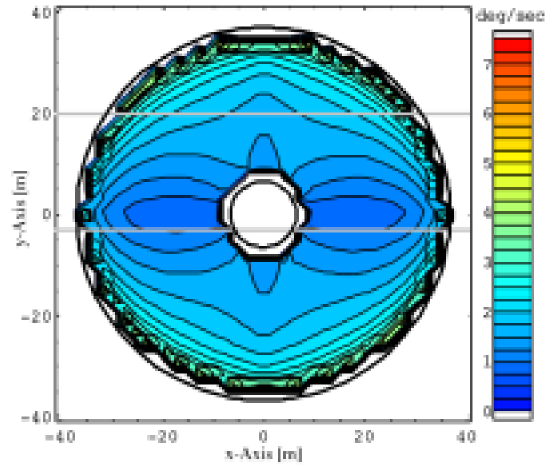


e) Workspace Plot Showing Joint and Crane Rate Limitations

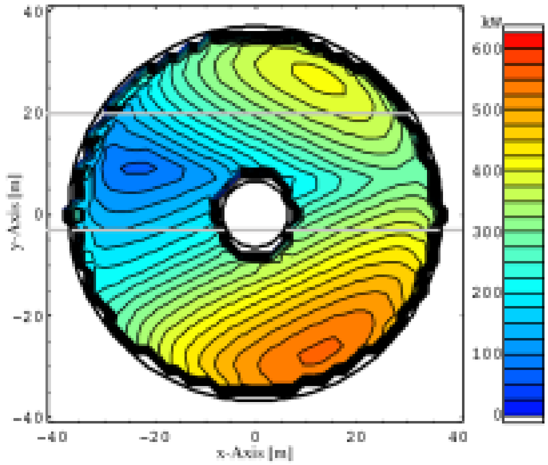
Figure 6.12 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a TACS Self-Leveling PCS Crane



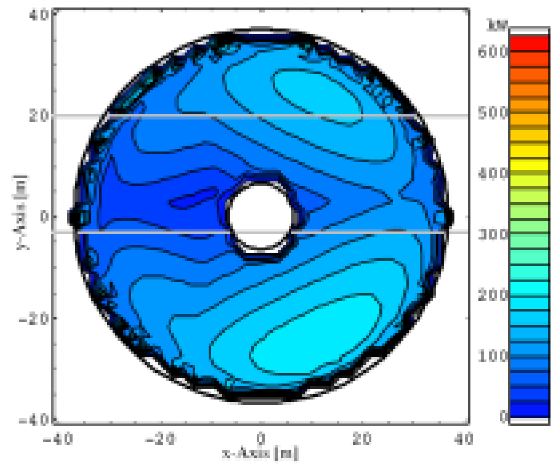
a) Maximum Slew Rates



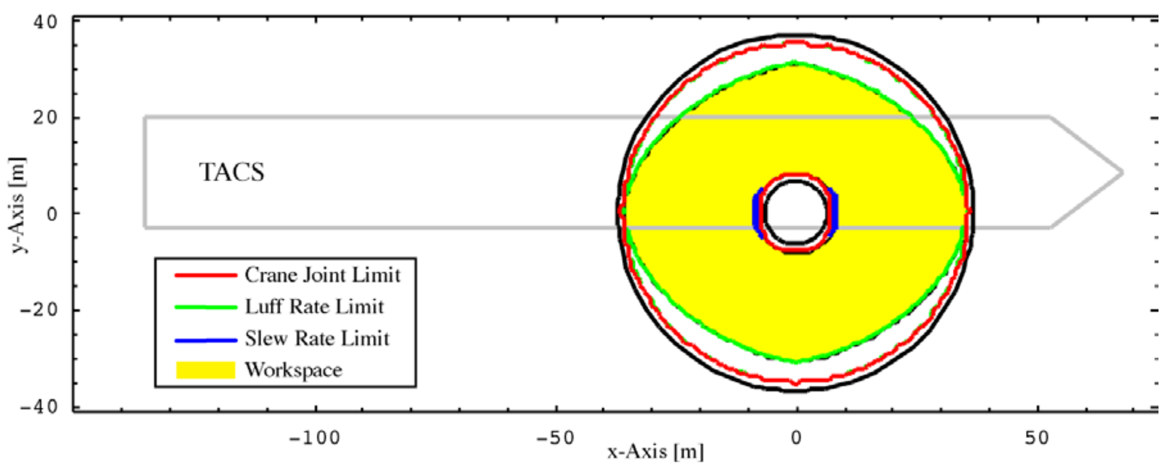
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

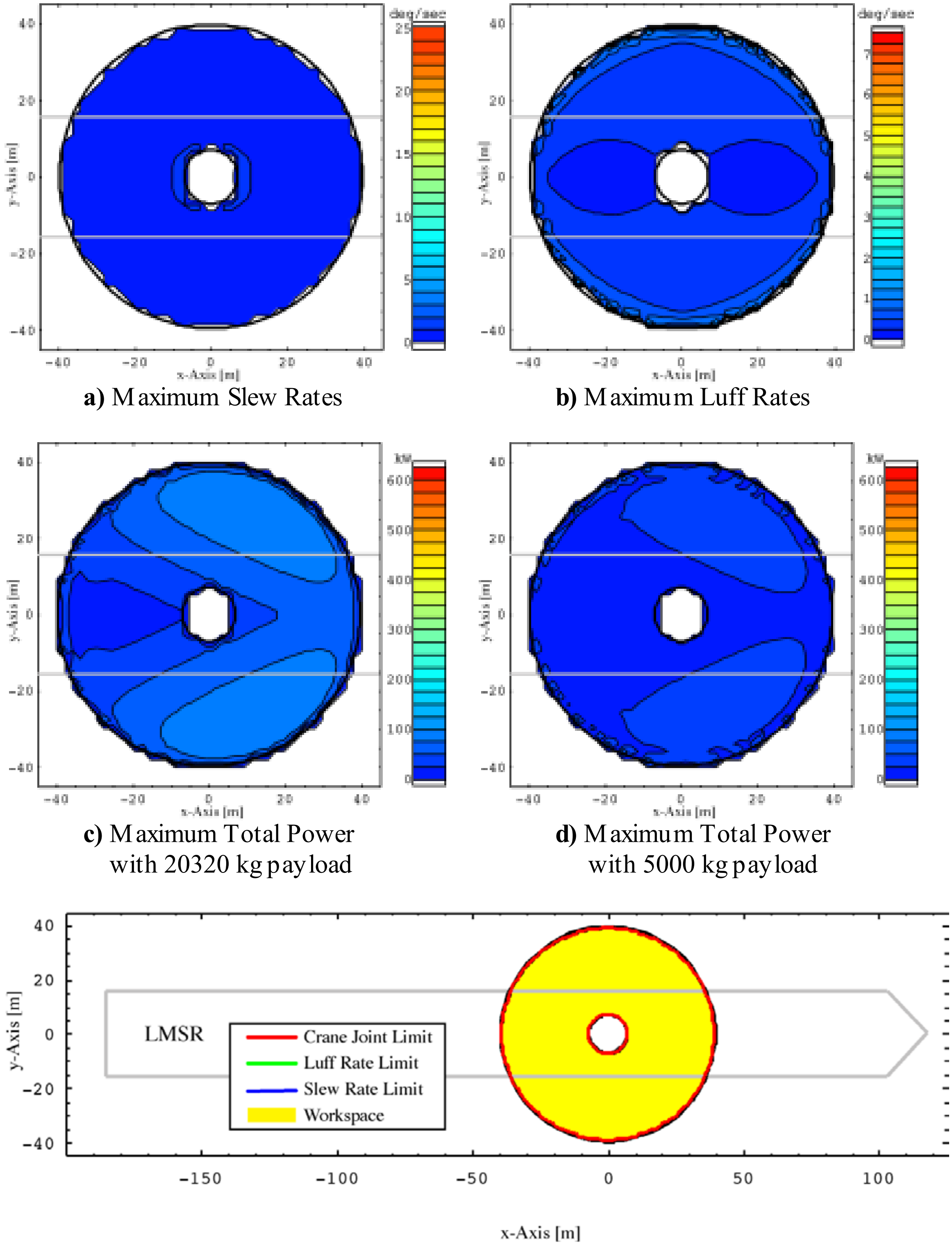


d) Maximum Total Power with 5000 kg payload



e) Workspace Plot Showing Joint and Crane Rate Limitations

Figure 6.13 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a TACS Self-Leveling PCS Crane



a) Maximum Slew Rates

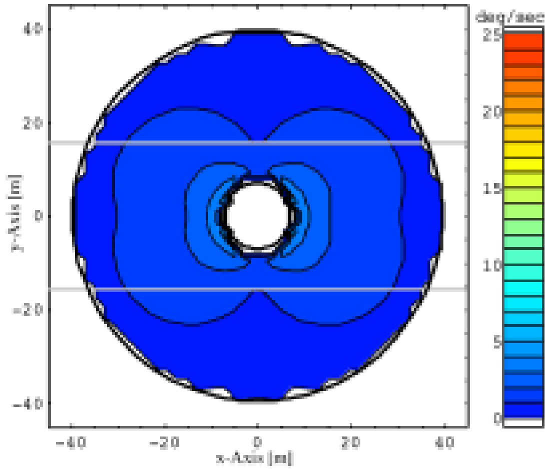
b) Maximum Luff Rates

c) Maximum Total Power with 20320 kg payload

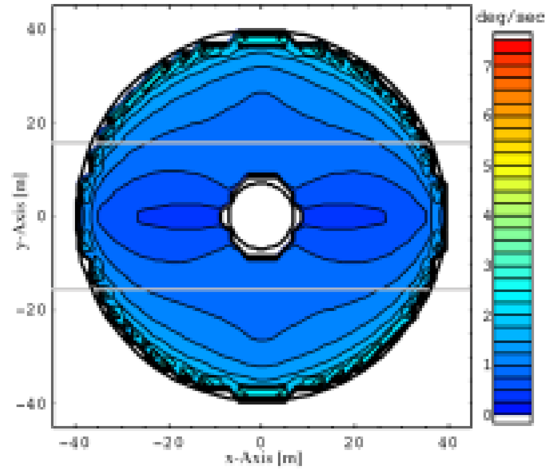
d) Maximum Total Power with 5000 kg payload

e) Workspace Plot Showing Joint and Crane Rate Limitations

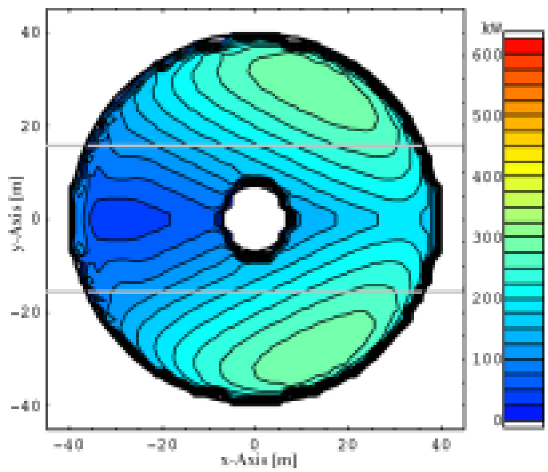
Figure 6.14 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a LMSR Self-Leveling PCS Crane



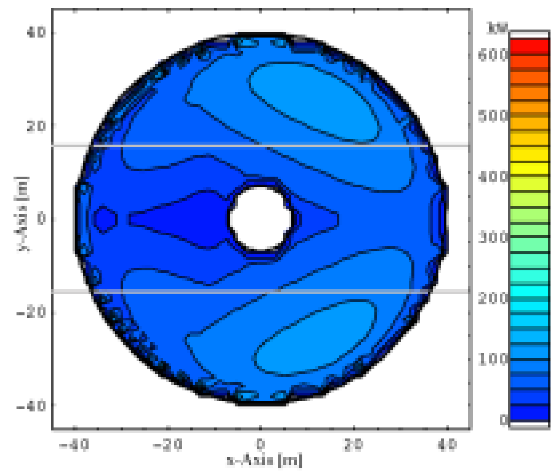
a) Maximum Slew Rates



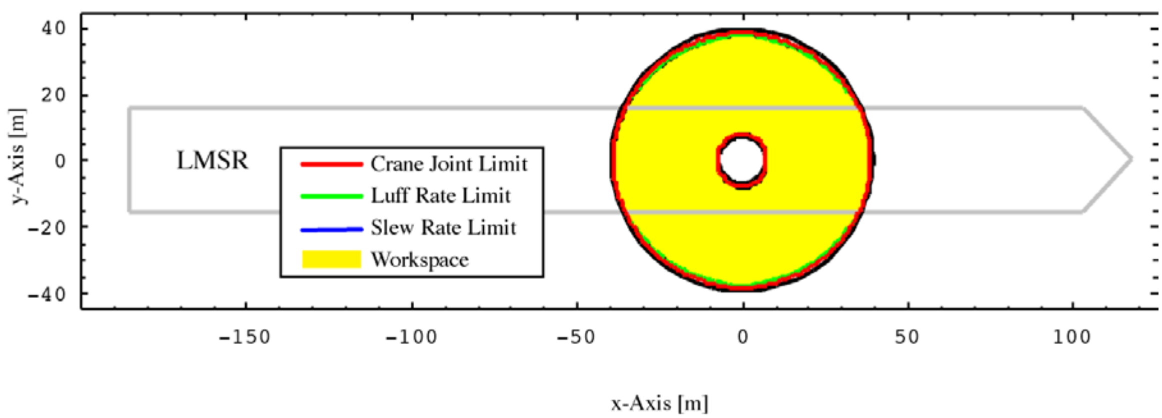
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

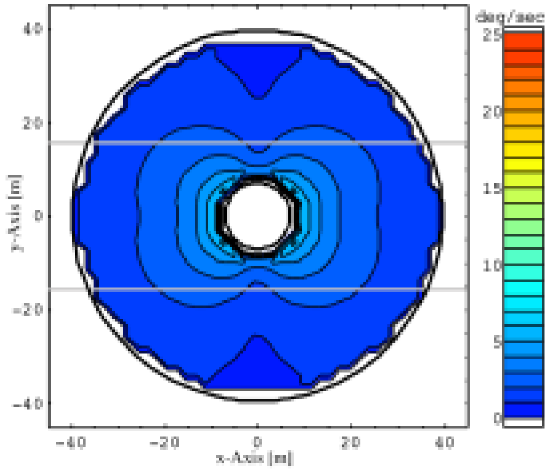


d) Maximum Total Power with 5000 kg payload

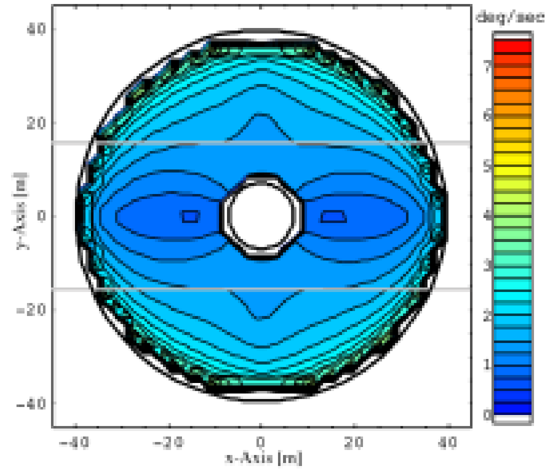


e) Workspace Plot Showing Joint and Crane Rate Limitations

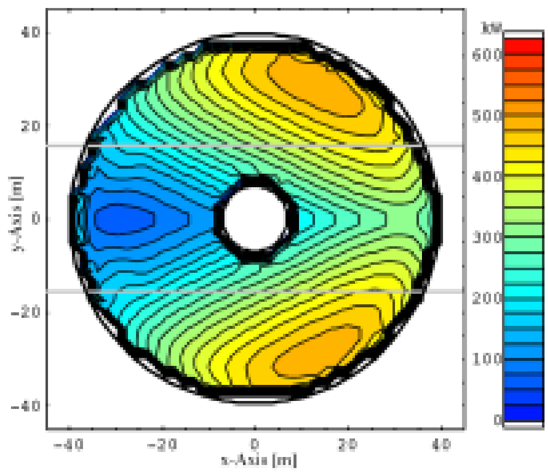
Figure 6.15 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a LMSR Self-Leveling PCS Crane



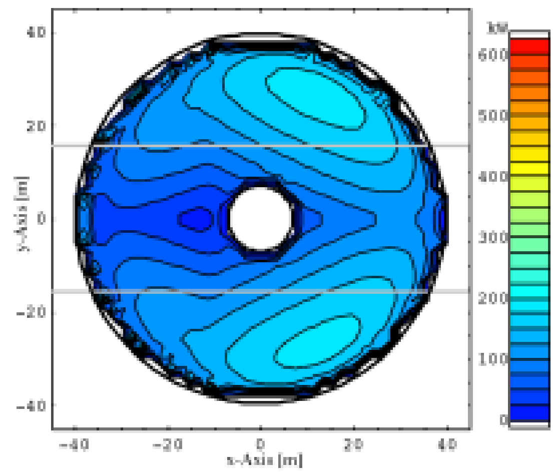
a) Maximum Slew Rates



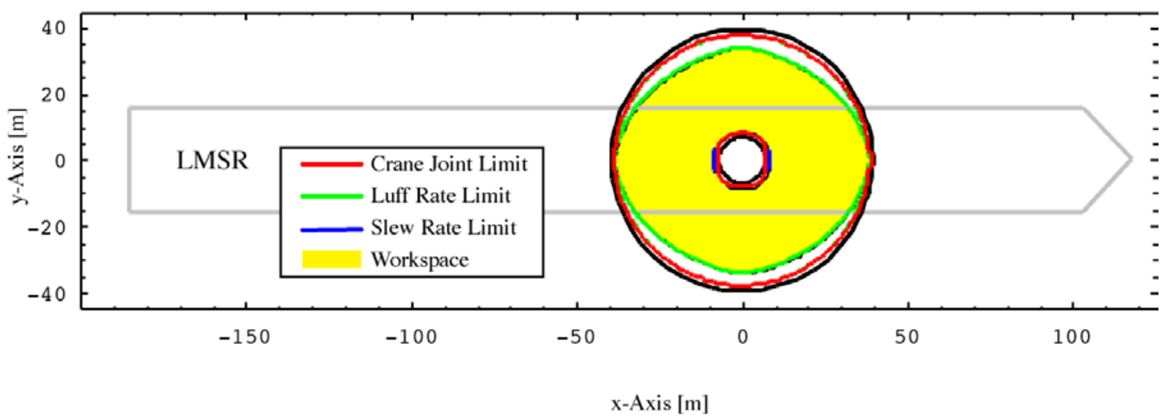
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload



d) Maximum Total Power with 5000 kg payload



e) Workspace Plot Showing Joint and Crane Rate Limitations

Figure 6.16 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a LMSR Self-Leveling PCS Crane

6.1.4.4. Active Target Tracking

This section describes a control strategy that would allow a shipboard crane payload to precisely track a target area on the receiving ship. The payload will nominally hang down from the boom tip along the gravity vector. A simple target tracking strategy might be to have the boom tip track the target point and hope that this inertial boom tip motion will not cause too much payload swing. However, since the natural frequency of the ships involved is close to the resonant frequency of the payload spherical pendulum, this strategy would result in a near resonant growth of payload swing.

Instead of just controlling a certain boom tip motion, it is possible to track a target with a spherical pendulum by controlling both the horizontal boom tip motion and the resulting required payload swing. The boom tip motion must be of a specific type and phase angle relative to the target motion such that the payload swing will result in the payload always being above the target point. This principle is illustrated in the simplified cart-pendulum system in Figure 6.17 below. Let y represent the target ship motion and x represent the boom tip motion. The boom tip motion x and the swing angle θ must be controlled such that the final payload position will track the desired target position y.

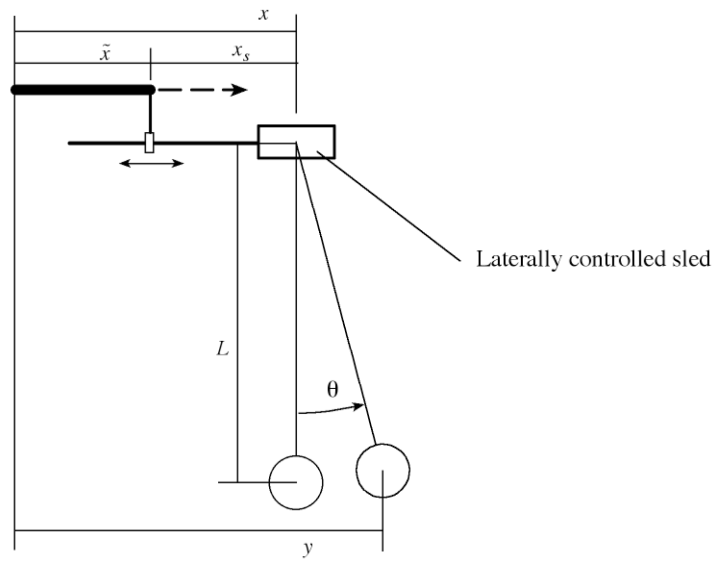


Figure 6.17 - Simplified Crane Model Illustrating the Target Tracking Control Strategy

The control strategy to achieve this effect depends directly on the horizontal target motion, the horizontal target acceleration, as well as the payload natural frequency. First, a PCS control strategy decouples the crane ship motion from the target ship motion. Second, specific boom tip corrections are computed that will result in the required payload motion to track an arbitrary target. Note that the target motion can have an arbitrary continuous trajectory and is not required to be sinusoidal.

Since this strategy requires both target motion and target acceleration, as well as the PCS sensor equipment, this control solution requires accurate measurements of the primary and secondary ship motion, as well the payload swing angles.

The size of the required boom tip corrections to track a target motion depends on the ratio between the target motion frequency and the natural payload pendulation frequency. If this ratio is small, then the target is moving much slower than the natural pendulation of the payload. In this case the control solution essentially has the boom tip follow the target motion. If this ratio is large, then the target motion is moving much faster than the natural payload pendulation frequency. In this case the boom tip motions will become very large to pull the payload along. Generally speaking, faster frequency or simply high accelerations will directly cause large boom motions. However, with the given natural ship rotation periods of about 10-15 seconds, ratios between target motion frequencies and payload frequencies are close to 1. If the target is moving at the natural frequency of the payload pendulum, then the control strategy requires no boom tip corrections. Instead, the target is tracked by the natural pendulation of the payload. How large this pendulation will need to be depends on the amplitude of the target acceleration. With the near-unit frequency ratio for the skin-to-skin problem, this results in minimal crane joint motions to provide the required boom tip motion. Note that it is assumed here that the required payload swing angle and phase have already been achieved. The transient control requirements to achieve this steady-state boom tip and payload swing motion have not been considered in this study.

Ideally this control strategy would be able to perfectly track a target motion. Contrary to the PCS and self-leveling PCS control studies, there is no need to add the target ship motions from Figure 6.1 to establish the payload tracking error.

The following plots show the TACS TG3637 crane rate and performance requirements to achieve steady state perfect tracking of the target motion. Transient rate and power requirements to produce the required swing angles are not considered. The skin-to-skin scenario has the vessels 6 meters apart. If the payload is over the primary crane ship, this control shows the crane requirements to track the crane ship deck. If the payload is more than 6 meter port or starboard, then the crane requirements are shown to track a ship undergoing the same roll and pitch amplitudes, but out of phase to provide the worst case result. This is why these contour and workspace plots appear segmented into three zones. Six plots are shown for the three ship motion cases with the standard and self-leveling TACS PCS crane. The rate and performance requirements are only slightly higher than the un-tracked control strategy (PCS) due to the near unit frequency ratio. Running simulations for the LMSR case provide similar results, which are not shown in this report.

Thus, with the active target tracking control strategy it is possible to have the payload track a target ship motion. Once tracking a target (i.e. ignoring the transient effect) only a small increase in the crane joint rate and power commands are observed. However, several practical hurdles remain to make such a solution feasible. Since the control depends directly on the target acceleration, any rapid motions of the target ship ,due to wave interactions, fenders, etc, will result in very large required joint rates and joint motions due to the frequency ratio becoming much larger than one. These large rates may saturate the crane servo limits and result in large tracking errors.

Further, this control strategy can require large payload swing relative to the gravity vector. The magnitude of the swing angle depends directly on the target motion acceleration. For the steady-state sinusoidal ship roll and pitch motions, Figure 19 shows the resulting required swing angle. As the target ship experience faster frequency content than the natural periods of 12 and 13 seconds, the swing angles will grow proportionally larger with the increased target accelerations. The resulting large swing angles could pose structural concerns since the boom is designed to ideally carry the payload weight in compression.

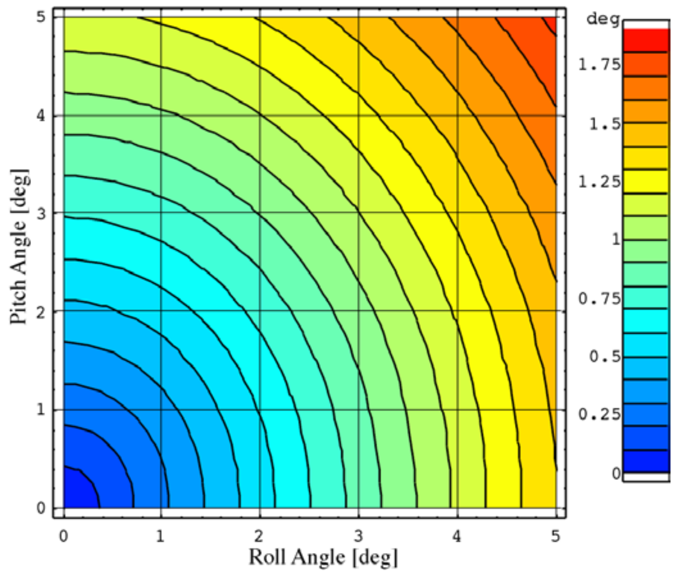
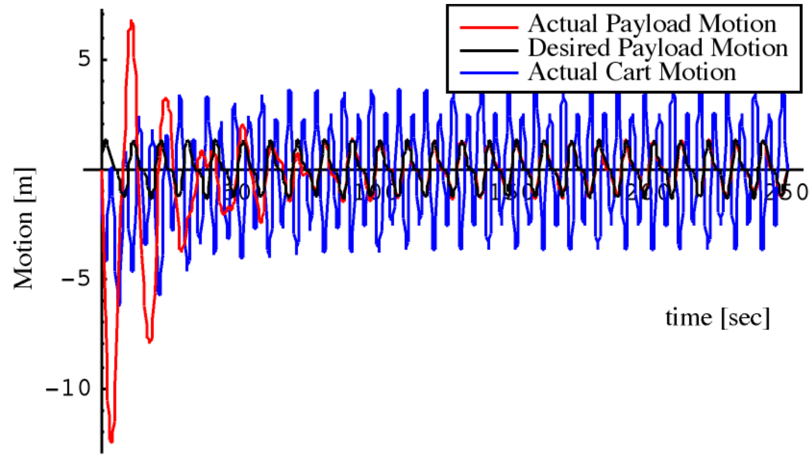
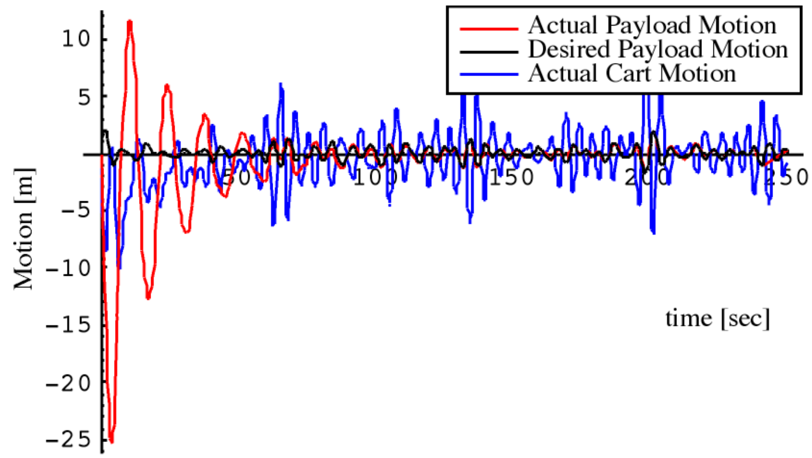


Figure 6.18 - Required Payload Swing Angles for Various Steady-State Target Ship Motion Amplitudes

Another control issue that must be considered is that the transients crane motions to achieve the required swing may require non-trivial crane rates to achieve. Figure 6.19 illustrates how the simplified 1-D cart-pendulum crane dynamic model in Figure 6.17 is able to track a general target motion. The hoist length is set to 35 meters. The target motion *y* has an amplitude of about 1 meter. Two target motion frequency content cases are shown. The first case has a simpler motion with only 0.1Hz and 0.2 Hz components. The second case has a more general motion with a spread of frequencies present between 0.1Hz and 0.2Hz. Note that these frequencies are faster than the typical natural crane ship motions. They illustrate how large the required boom motion (*x*-motion in this simple crane model) can become if large accelerations are present. In both simulations the payload is initially at rest. After the tracking control is engaged, the payload motion (red line) is made to asymptotically track the target motion (black line). This occurs for both the simpler and more complex target motion cases.



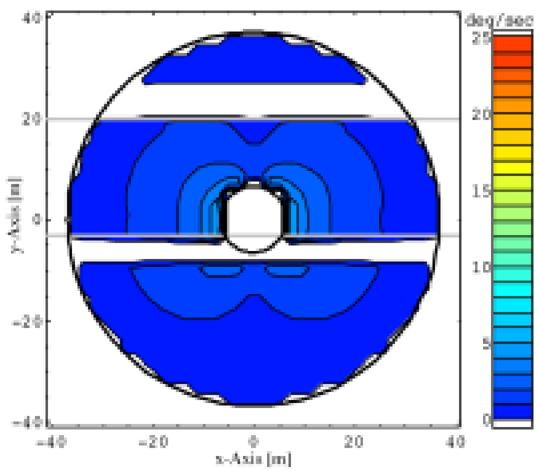
a) Target Motion Consists of 0.1Hz and 0.2Hz Signals



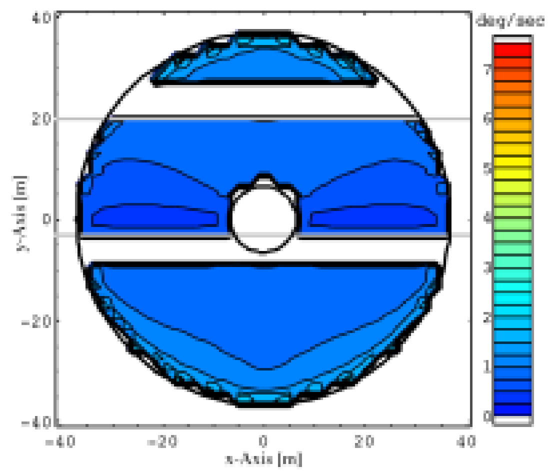
b) Target Motion Consists of a Spread of Frequencies between 0.1Hz and 0.2Hz

Figure 6.19 - Illustration of the Simplified Crane Model Tracking a General Target Motion

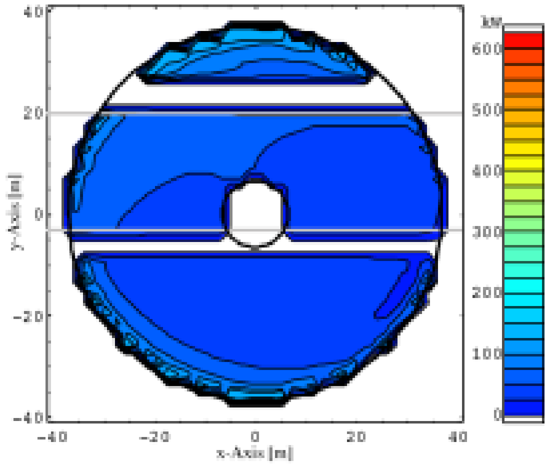
The active target tracking strategy shows great promises for the precise payload landing on target ships at open seas. However, it is a challenging research topic with many control and sensor related questions remaining to be answered. It does have the potential to provide very small payload tracking errors.



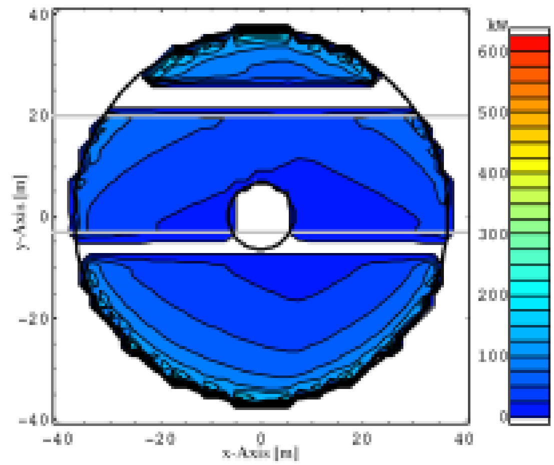
a) Maximum Slew Rates



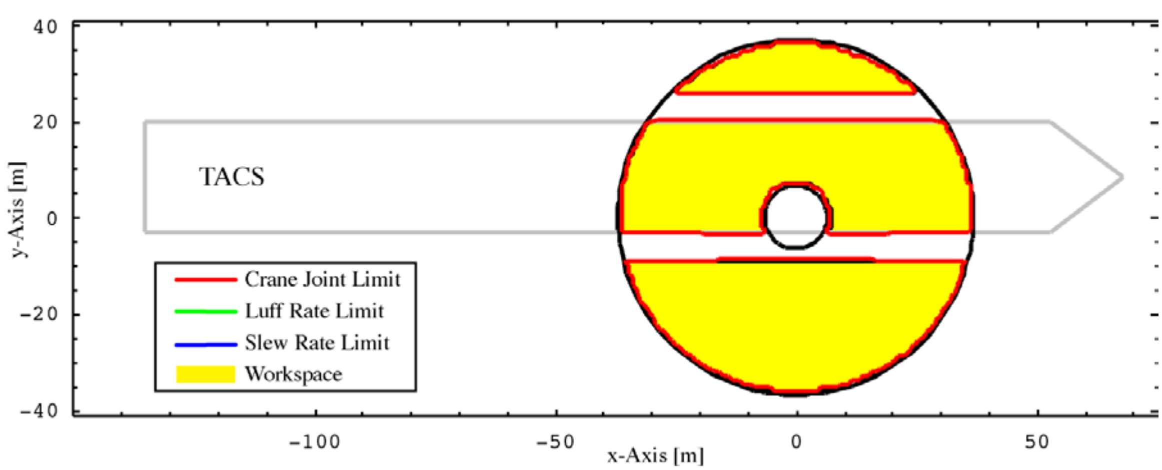
b) Maximum Luff Rates



c) Maximum Total Power
with 20320 kg payload

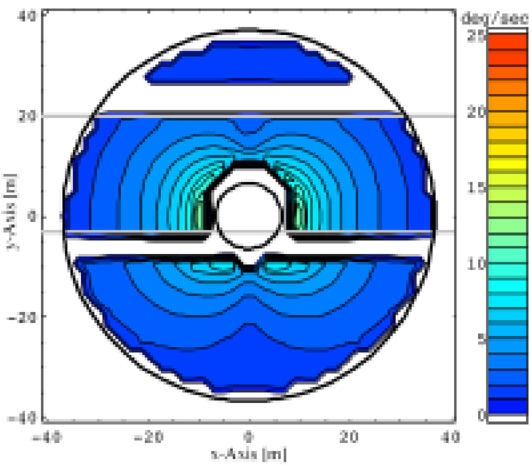


d) Maximum Total Power
with 5000 kg payload

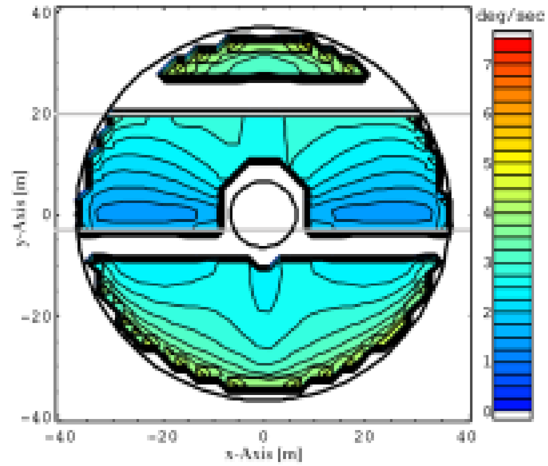


e) Workspace Plot Showing Joint and Crane Rate Limitations

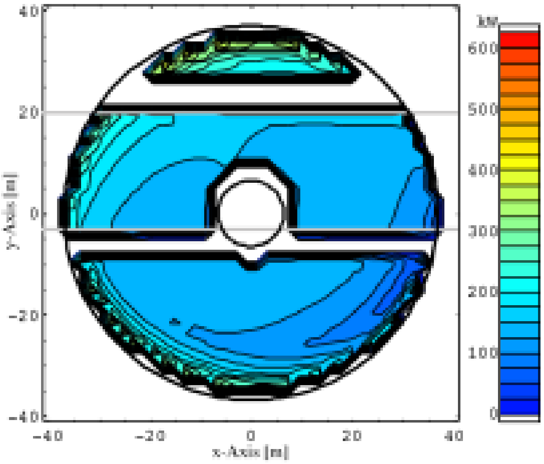
Figure 6.20 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a TACS PCS Crane with Target Tracking Active



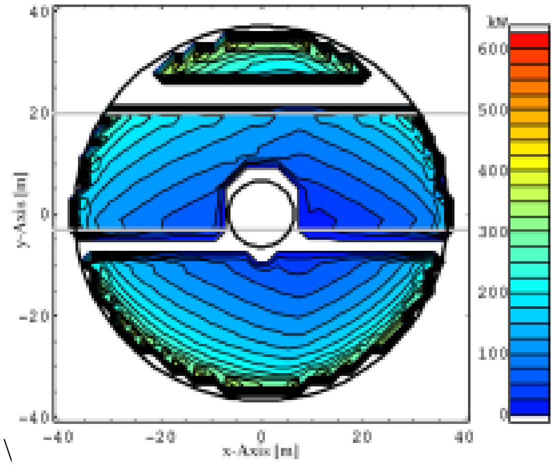
a) Maximum Slew Rates



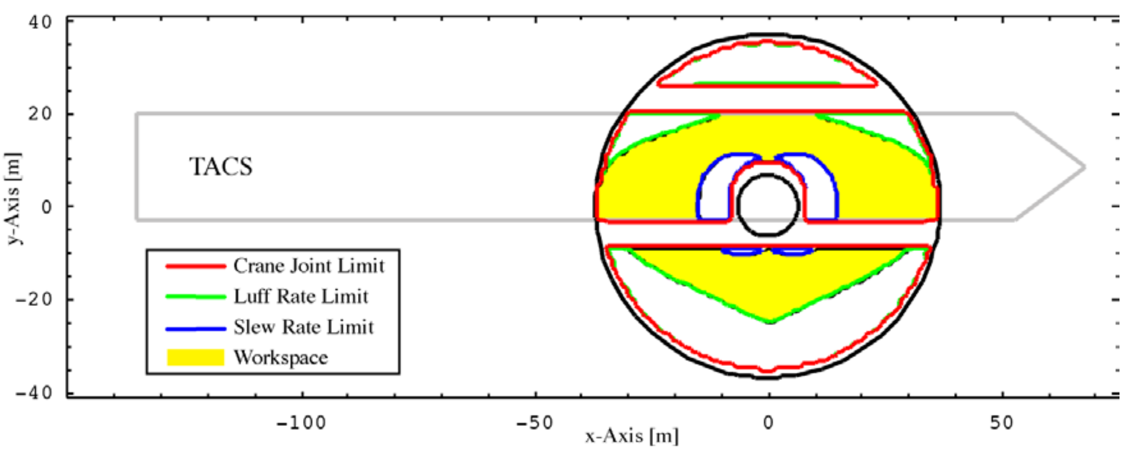
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

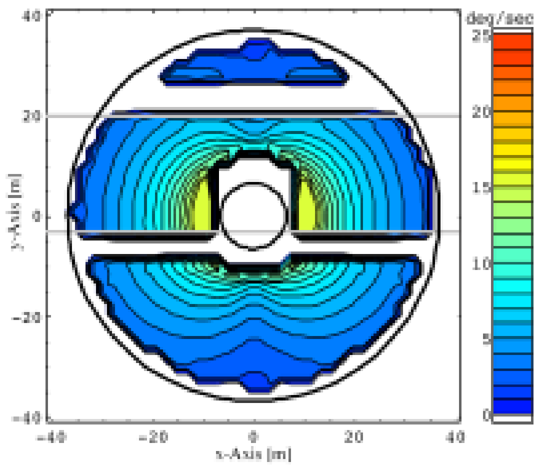


d) Maximum Total Power with 5000 kg payload

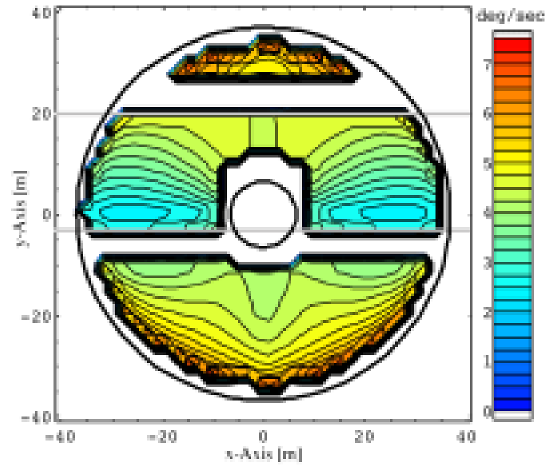


e) Workspace Plot Showing Joint and Crane Rate Limitations

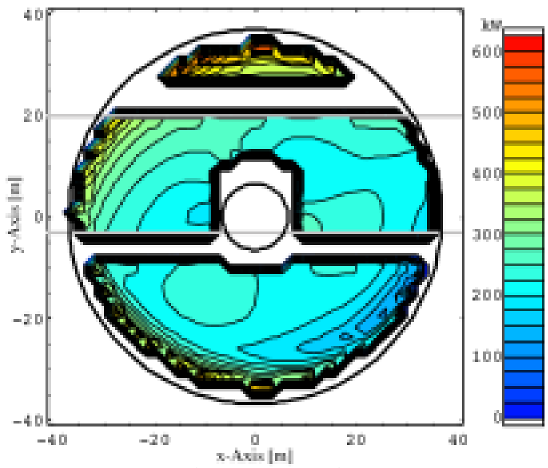
Figure 6.21 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a TACS PCS Crane with Target Tracking Active



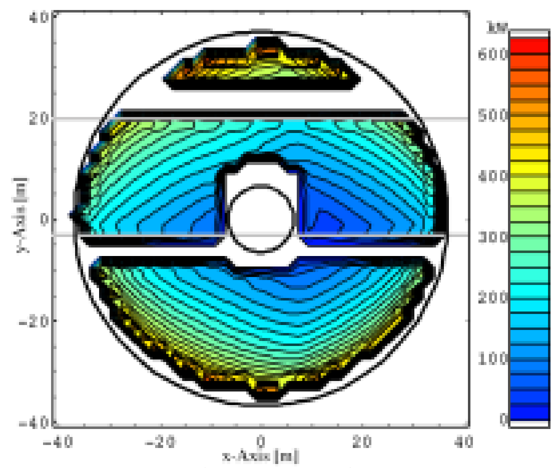
a) Maximum Slew Rates



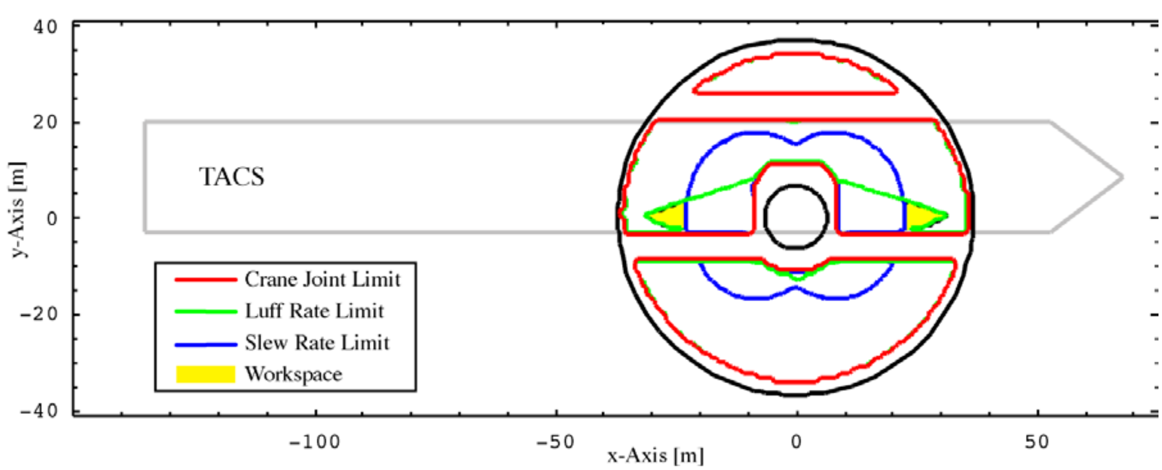
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

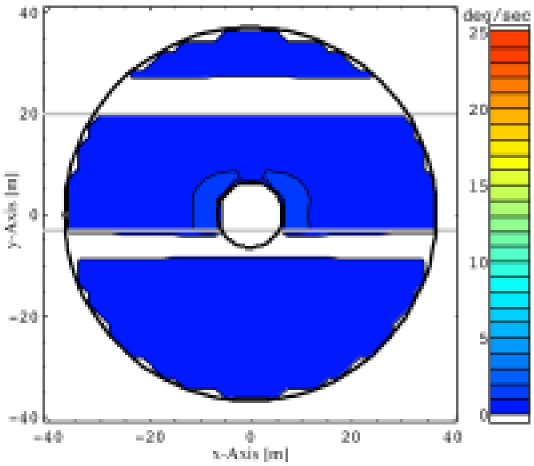


d) Maximum Total Power with 5000 kg payload

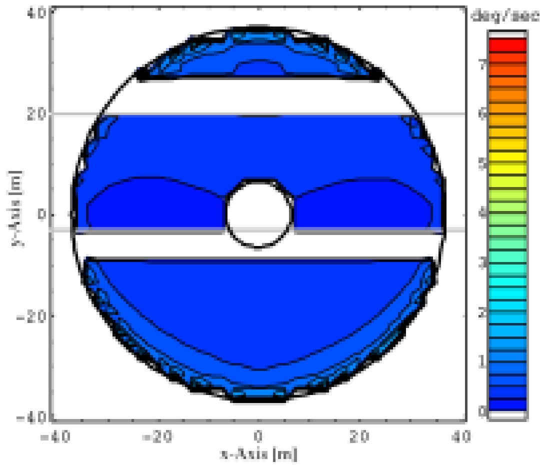


e) Workspace Plot Showing Joint and Crane Rate Limitations

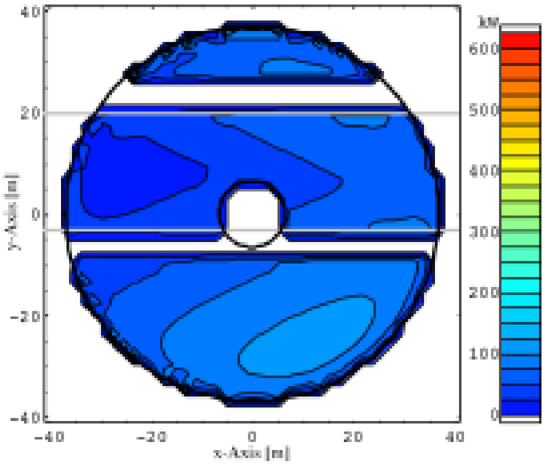
Figure 6.22 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a TACS PCS Crane with Target Tracking Active



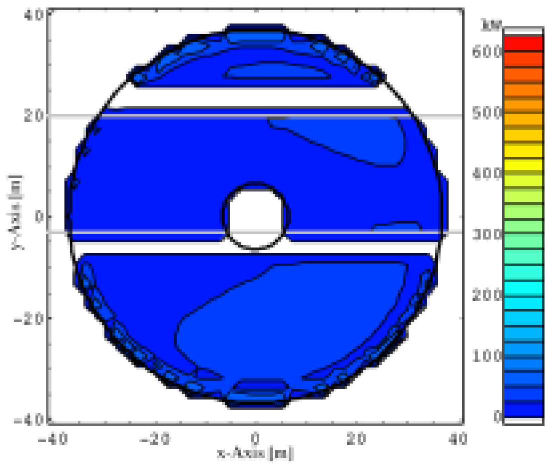
a) Maximum Slew Rates



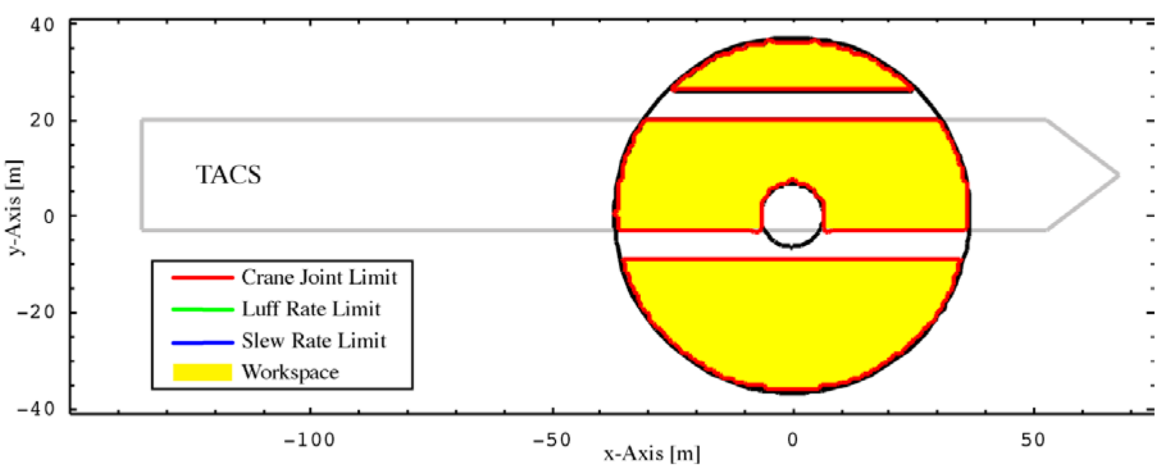
b) Maximum Luff Rates



c) Maximum Total Power
with 20320 kg payload

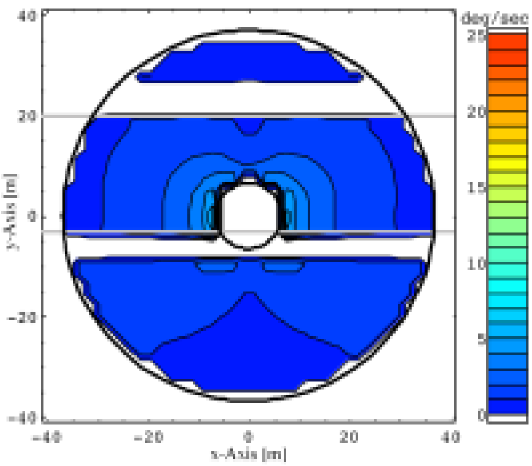


d) Maximum Total Power
with 5000 kg payload

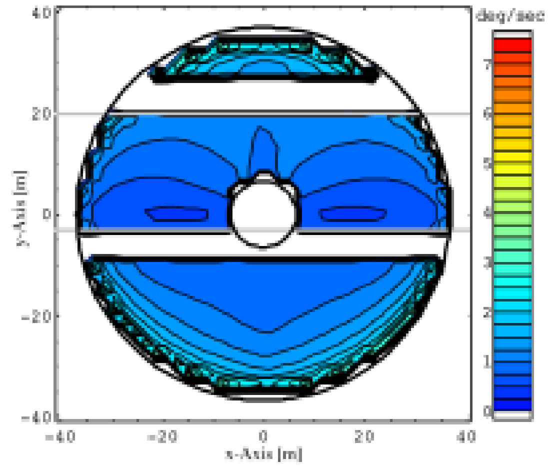


e) Workspace Plot Showing Joint and Crane Rate Limitations

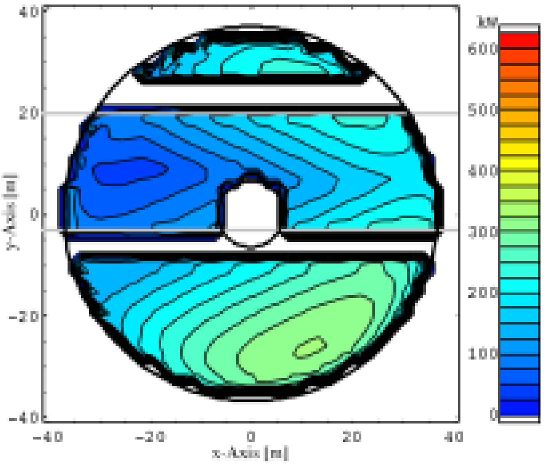
Figure 6.23 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 1 with a TACS Self-Leveling PCS Crane with Target Tracking Active



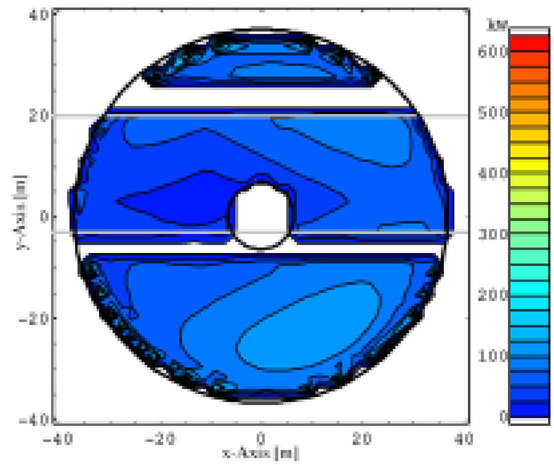
a) Maximum Slew Rates



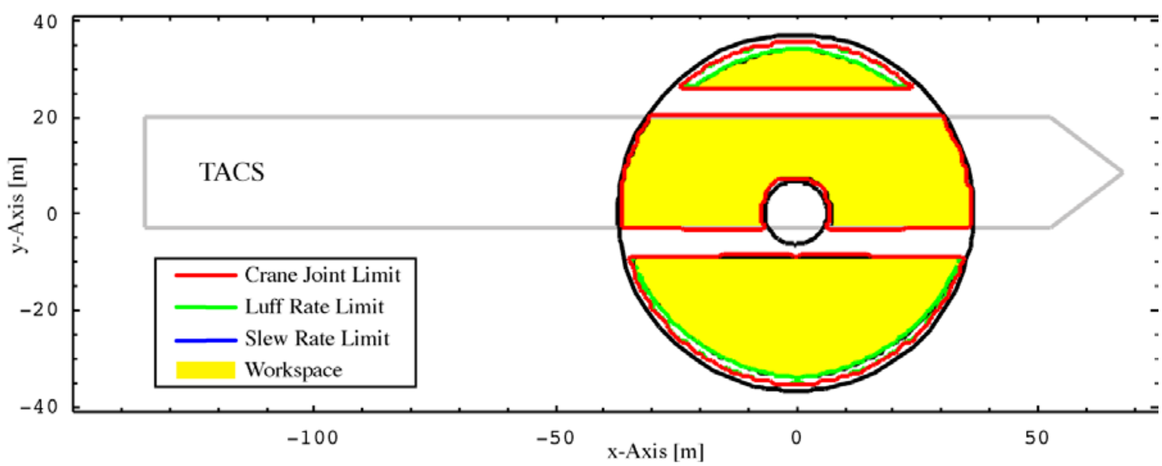
b) Maximum Luff Rates



c) Maximum Total Power with 20320 kg payload

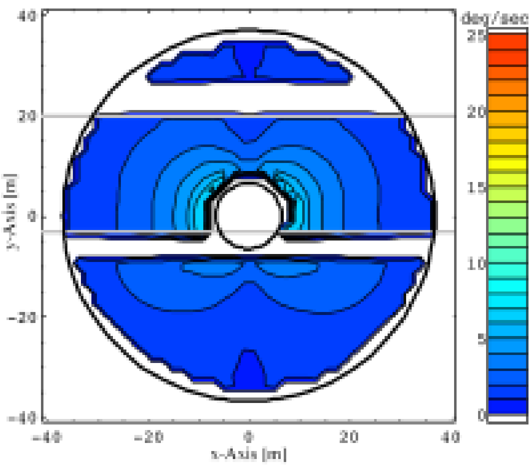


d) Maximum Total Power with 5000 kg payload

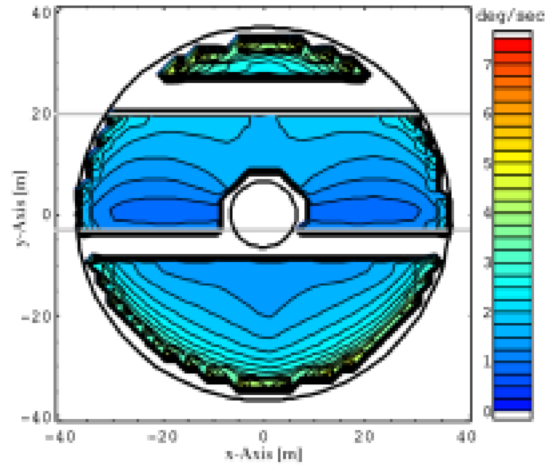


e) Workspace Plot Showing Joint and Crane Rate Limitations

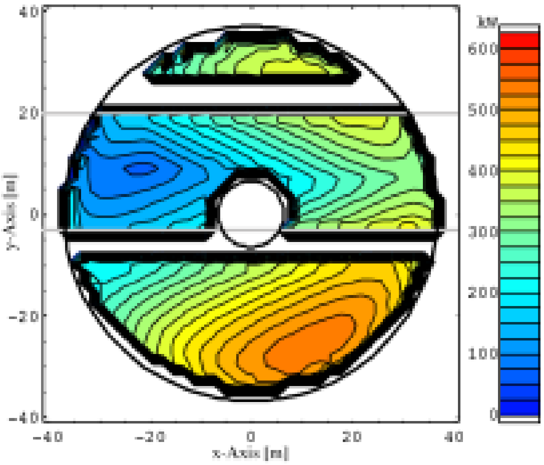
Figure 6.24 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 2 with a TACS Self-Leveling PCS Crane with Target Tracking Active



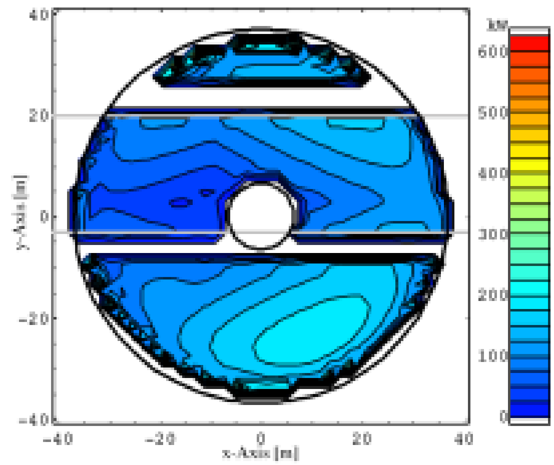
a) Maximum Slew Rates



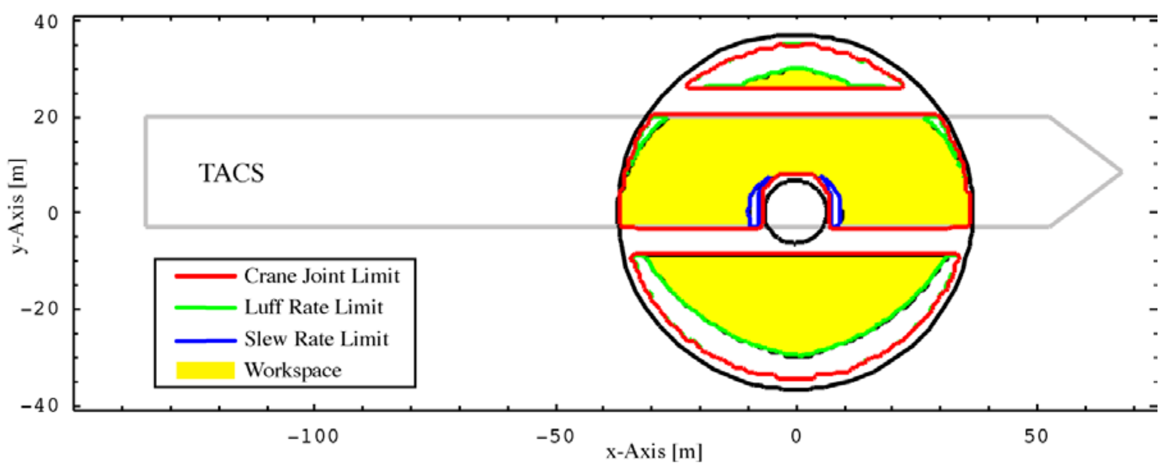
b) Maximum Luff Rates



c) Maximum Total Power
with 20320 kg payload



d) Maximum Total Power
with 5000 kg payload



e) Workspace Plot Showing Joint and Crane Rate Limitations

Figure 6.25 - Crane Performance Requirements and Potential Workspace Illustration for Ship Motion Case 3 with a TACS Self-Leveling PCS Crane with Target Tracking Active

6.1.5. Summary

This section studied the effect of various degrees of ship motion on the skin-to-skin cargo transfer problem. The ship roll and pitch motions were assumed to vary between 1 and 5 degrees and occur at periods of 12-13 seconds. The RBTS system performance is provided as a benchmark solution. The RBTS system is already installed on many cranes. However, it only attempts to passively reduce the onset of payload swing by pulling in the rider block and shortening the effective payload hoist length and associated pendulation frequency. Numerical simulations illustrate the issues that a RBTS system will have operating at the different ship motion cases. Slack lines may occur which can drastically limit the operational workspace. Also, the payload is not kept inertially fixed and the ship roll and pitch induced translation can cause significant payload tracking errors relative to the target ship.

Employing a pendulation control system results in the payload motion being decoupled from the crane ship motion. Tracking errors are substantially reduced compared to the RBTS crane and the payload motion relative to the target ship is smoother and easier to predict by the crew. This should increase safety and facilitate cargo landing. However, for larger ship motion the crane joint rate requirement can grow rather large with this method, especially at workspace boundaries. To compensate for this, a self-leveling PCS crane is discussed. This crane has a new moving base installed that will compensate for the ship roll and pitch angles. The result is a significant reduction in crane joint rate requirements and increase in the operational workspace. However, implementing such a moving base would incur the cost of designing and installing the two degrees-of-freedom base on the crane.

Lastly, a future crane control system is discussed that would make the payload track the target ship motion perfectly. Since the target ship motion period is similar to that of the payload pendulation period, the crane joint rate increase to achieve the steady-state tracking is not very large. However, many control and sensor issues must be researched to make this a practical solution. This control strategy will require precise sensing of the target ship motion. Also, the required payload swing may cause structural concerns with the boom.

Ultimately, finding an appropriate crane design for the skin-to-skin cargo transfer problem depends on the amount of ship motion present and the cargo tracking error requirements. This report illustrates how the crane requirements will increase as the ship motions grow larger and/or the target tracking error requirements increase.

6.2. Gantry Cranes

Ship mounted gantry cranes are less common than pedestal cranes, but are suitable for some applications. They are designed for loading and unloading the ship's container holds to/from a pier. These cranes tend to have high hoisting and trolley speeds, which allow for very rapid cargo movement.

Shipboard gantry cranes come in two basic configurations. One is the type seen on the Waterman Class MPS and LASH ships, which have the cargo lifting trolley mounted on the underside of the moving gantry. The gantry is positioned directly above the cargo for lifting. The other type combines a movable gantry with a slewing boom. The slewing boom is mounted on the top of the gantry and can access cargo in a greater area than the standard gantry crane. The basic capabilities of both types are similar.

Load capacities are 30 LT and higher, and the reach outboard of the ship is in the 33' (10 m) range.

A key advantage to gantry cranes is that the design allows for inherently stable cargo movements, and also allows for load stabilizing systems, similar in concept to the Rider Block Tagline System, to be effectively incorporated. This is especially relevant to the SS5 skin-to-skin problem. Figure 6.26 shows the multicable lifting arrangement common on gantry cranes.

Figure 6.26 – Gantry Crane Lifting Arrangement

The paper "RoboCrane® for Portable Ports and Mobile Offshore Base"[6] details a system that utilizes a rider block system with taglines for additional load stabilization on a gantry style crane. A diagram of this system is shown in Figure 6.27.

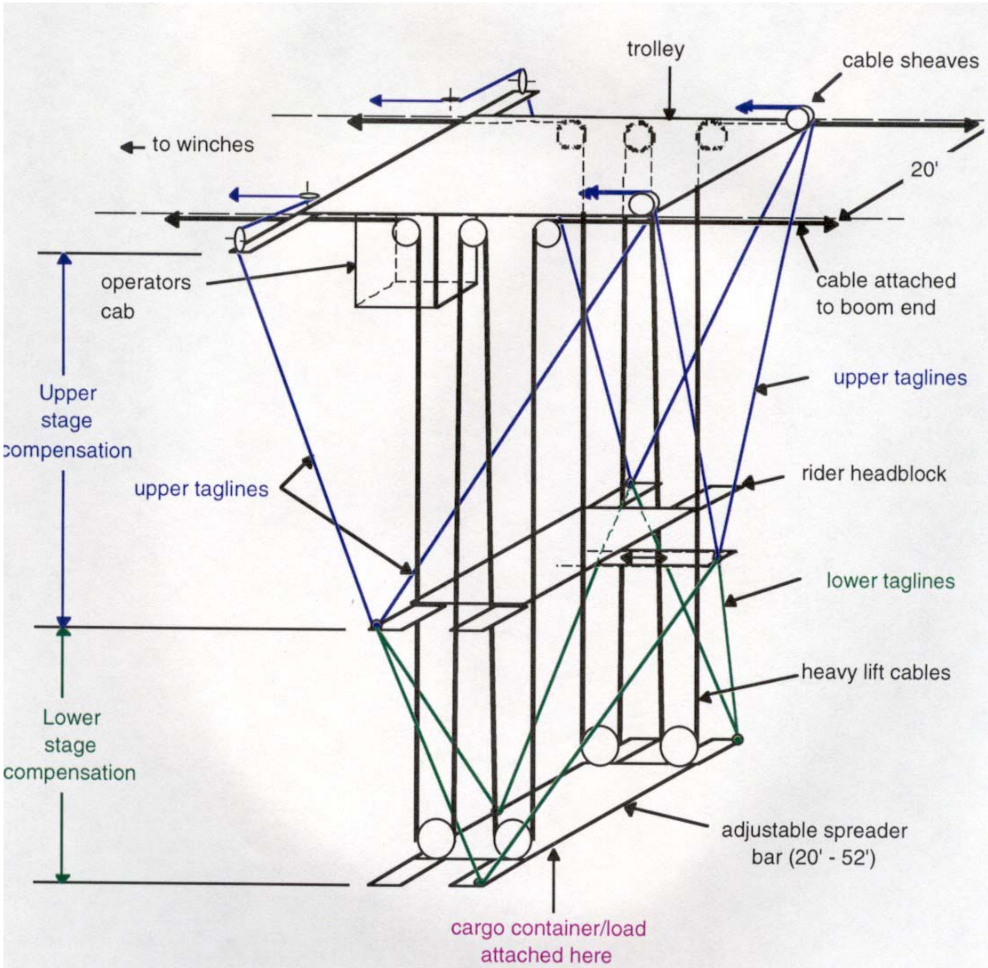


Figure 6.27 – Gantry Crane Load Stabilizing System

Another important feature of the gantry crane load stabilization design is that the available workspace of the crane is not reduced by the stabilization system, and is not necessarily adversely impacted by ship motions. Assuming the load stabilization is effective and the crane has a high hoist rate, safe and controlled cargo movement will be possible.

An important factor when considering a gantry crane design is it's large size and corresponding ship arrangements impact, compared to pedestal mounted boom cranes. The crane needs rails mounted on the deck that span the entire desired working area. This can also be viewed as an advantage however, in that a well designed system would allow a small number of cranes to service a large deck area, due to the gantry crane's mobility. Another limitation of current gantry crane designs, compared to boom cranes, is the limited outreach from the ship they provide. The aforementioned LASH and Waterman gantry cranes, as well as the container cranes mounted on the recently built refrigerated container ships for Dole, have a 33' (10 m) outreach. This is adequate for offloading to a pier or alongside lighterage, but could be insufficient for skin-to-skin operations.

In summary, gantry crane designs have a number of desirable features that warrant further investigation, and the limitations of the design are not so severe as to exclude them from consideration. When a better defined set of requirements are developed, based on motions and cargo requirements, a candidate platform should be identified and a detailed study performed to determine if a gantry crane can be designed to specifically designed to meet those requirements.

6.3. Rigid Arm Cranes

For the purposes of this report, two types of rigid arm cranes are considered: knuckle boom and telescoping boom cranes.

A knuckle boom crane, as shown in Figure 6.28, has a multipart boom with one or more rotating joints. These joints allow for a greater variety of movements than on a standard boom crane. Of particular interest in the skin-to-skin scenario is the ability to lower the tip of the boom without extending the reach of the crane. This will enable cargo to be picked up and placed down with very short hoist cable lengths. In addition to the greater boom agility, standard practice with these cranes is to hoist cargo all the way to the boom tip. These two features minimize cargo pendulation and enhance the operators' ability to quickly, safely, and accurately place loads.

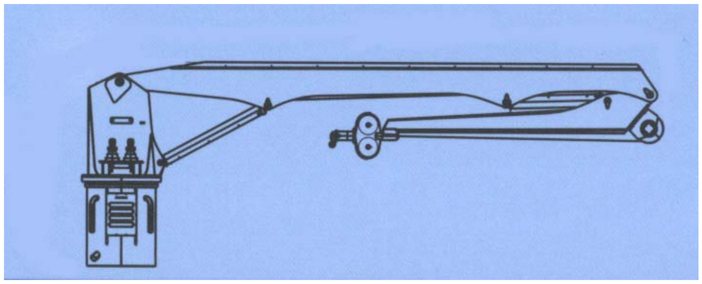


Figure 6.28 – Knuckle Boom Crane

The major limitation of knuckle boom cranes is their low payload capacity, when compared to standard boom and gantry cranes. The PKM 1200 knuckle boom crane from CranePower, now the manufacturer of Palfinger marine cranes, has a capacity of 6 LT at approximately 65'. This is clearly far short of the 30 LT capacity seen in container capable cranes. However, for scenarios where 6 LT is a sufficient capacity, such as an augmentation of the current and future STREAM UNREP systems, this type of crane could be ideally suited.

Telescoping boom cranes (Figure 6.29) have variable length booms that extend along a single axis. They provide the same capability to lift the load up to the boom tip as a knuckle boom crane, without the increased agility provided by the rotating joint. The advantage to the telescoping boom design is that greater payload capacities and reaches can be attained.

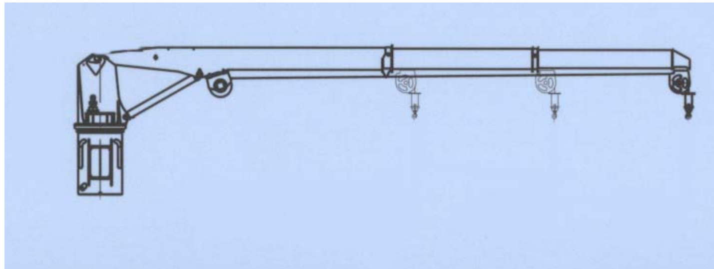


Figure 6.29 – Telescoping Boom Crane

These cranes normally are rated for a certain load capacity at certain reaches. The PTM 1700 crane from CranePower is rated at 17 LT at 31', 9 LT at 52', and 5.7 LT at 72'. Like the knuckle boom design, this type of crane would be well suited for moving smaller loads at a high rate during a skin-to-skin operation.

6.4. Trans-ship Bridge

An obvious approach to transferring cargo between vessels moored skin-to-skin would be to use a bridge, conveyor, or other rigid platform that could quickly move loads from one ship to the other. Such a system could potentially be used to transfer vehicles, large cargo such as containers, or small pallets. There are several issues that must be considered to assess the feasibility of such a system:

- Ship motion condition
- Ship size differences
- Ship to ship interface

The most obvious issue is the fact that each ship will be moving in 6 degrees of freedom. Some of these motions will be very small, but not necessarily insignificant. The larger motions, roll and pitch, will cause significant deflections of both ends of the ramp. These deflections will induce a twisting motion in the ramp, which as experienced in JLOTS operations with RO/RO ramps and RO/RO Discharge Facilities is a serious and difficult to solve problem. In addition to twisting the ramp, roll motions in particular will constantly change the linear distance the ramp must span. Any bridging system would have to be designed to withstand and/or compensate for these motions. This could involve multiple active motion compensation platforms, a telescoping ramp design, ball and socket type connection points, or another mitigation device.

The next issue that needs to be addressed is the wide variety of ship sizes that will be involved. Cargo will likely be transferred from one ship's deck to the other. In cases where the ships decks are not at the same height, another significant design challenge is presented. Now the ramp not only needs to be long enough to span the ship separation distance, but also to make up the height difference. The deck height difference between an LMSR and a T-ACS for example can be as much as 20 feet depending on load condition. This would increase the required length of the ramp by approximately 40%. Also, the ramp would be at a 45° angle, greatly increasing the requirements on any securing or cargo moving devices on the ramp.

Another issue is the interface between the two ships. In the case that the hypothetical ramp is on a fixed location on one ship, it's location would dictate the position where the receiving ship would be moored. Certain ships have limited areas for receiving cargo, and a ramp would necessarily require that location to be brought alongside, unlike a cargo crane with a significant reach that can service a greater area. A potential solution to this would be a slewing ramp design or a portable ramp, both features that would significantly increase design complexity.

One limited application were such a system may be feasible would be for a small pallet conveyor. If a receiving system was developed where this conveyor did not have to land or attach to the receiving ship, the system could prove useful. It would address some of the problems discussed above. However, the difficulties posed by the ship's size differences or interface issues may still exist.

6.5. Integrated loading platform

The ILP concept is very similar to the previously described trans-ship bridge/spanning ramp concepts. However, the primary difference lies in the fact that while the spanning ramp transfers cargo using the main deck area, an ILP consists of a retractable platform/ramp attached to the side of a sealift ship. This platform would have direct access to the main cargo holds and would allow rolling stock or cargo to be transferred directly in or out of the sealift ship. Figures 6.30 and 6.31 depict an early ILP concept and are based on an LMSR.

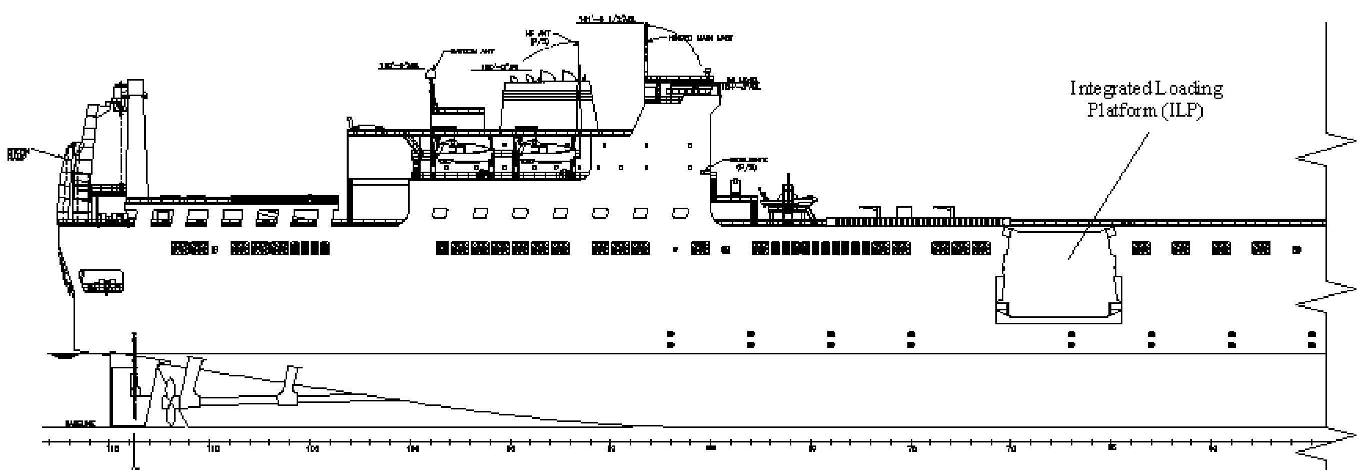


Figure 6.30 – Profile of LMSR with ILP

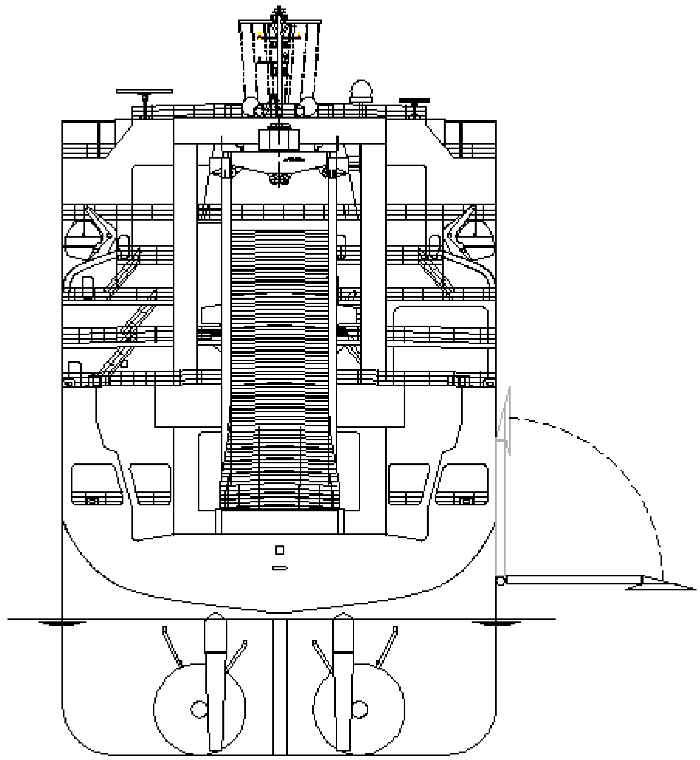


Figure 6.31 – Stern view of LMSR with deploying ILP

Te issues and concerns with the ILP are also similar to those mentioned in the previous section. Some additional issues include:

- Feasibility/cost of ILP installation on existing ships
- Platform weight and affect on ship stability

While these concepts may become a viable option if considered during the design stage of the MPF(F) or seabase, the technical and operational problems posed by a trans-ship bridge or ILP concept appear too great to recommend any further development at this time. Both concepts would require a landing area on the receiving ship large enough to accommodate the footprint of the system as well as compensate for the relative motion and additional stress once attached.

6.6. Liquid cargo transfer

The transfer of liquid cargo such as fuel in SS5 skin-to-skin connected replenishment operations will be required in order to sustain long-term operational capability. Current fueling-at-sea, or FAS, transfers generally make use of a standard 7" hose. However, hose size and the resulting pump rate can vary depending on the delivery and receiving ship. The following table summarizes potential ships along with their respective hose sizes and pump rates, taken from Naval Warfare Publication 14, Rev E, Replenishment at Sea.

Table 6.2 – FAS Transfer Table

FUEL	DELIVERY SHIP	RECEIVING SHIP	AVAILABLE HOSE SIZE IN INCHES (mm)	DESIGN PUMPING CAPABILITY GAL (m3)/HR/HOSE
F76	All AOs, T-AOs, AOEs, and AORs	All Types	7 (177.8)	180,000* (681.3)
	Carriers	Destroyers and Frigates	7 (177.8)	180,000* (681.3)
	AE-21 Class AE-26 Class AFS-1, 2, 3, 5 AFS-4, 6, 7	All Types Except Carriers	7(177.8) 7 (177.8) 7 (177.8) 7 (177.8) 7 (177.8)	40,000 (151.4) 45,000 (170.3) 42,000 (159.0) 60,000 (227.1)
	LHA-1 Class LHD-1 Class LKA-113 Class LPH-2 Class LPD-1 Class LPD-4 Class LSD-28 Class LSD-36 Class LSD-4a Class	Destroyers, Frigates, and Small Auxiliaries	7 (177.8) 7 (177.8) 7 (177.8) 6 (152.4) 6 (152.4) 4 (101.6) 6 (152.4) 6 (152.4) 2-1/2 (63.5)	135,000 (511.0) 135,000 (511.0) 135,000 (511.0) 120,000 (454.2) 60,000 (227.1) 60,000 (227.1) 36,000 (136.3) 60,000 (227.1) 15,000 (56.8)
F44	All AOs, T-AOs, AOEs, and AORs	Carriers Destroyers, Frigates, and Auxliaries Capable of Refueling Helicopters	7 (177.8) 2-1/2 (63.5)	180,000* (681.3) 15,000 (56.8)
	Carriers	Frigates	7 (177.8)	180,000* (681.3)

*Rate shown is with fueling probe. For Robb coupling or pigtail, the rate will be less.

Current UNREP operations generally conduct fuel transfer at speeds of 12 to 16 knots and at a standoff distance of 150 to 180 feet. In addition, U.S. ships normally transfer fuel by STREAM rig, by spanwire rig, bye close-in rig, or by a spanline rig. The STREAM rig is generally preferred in UNREP operations because it allows greater ship separation. An example of a STREAM rig is shown in Figure 6.32.

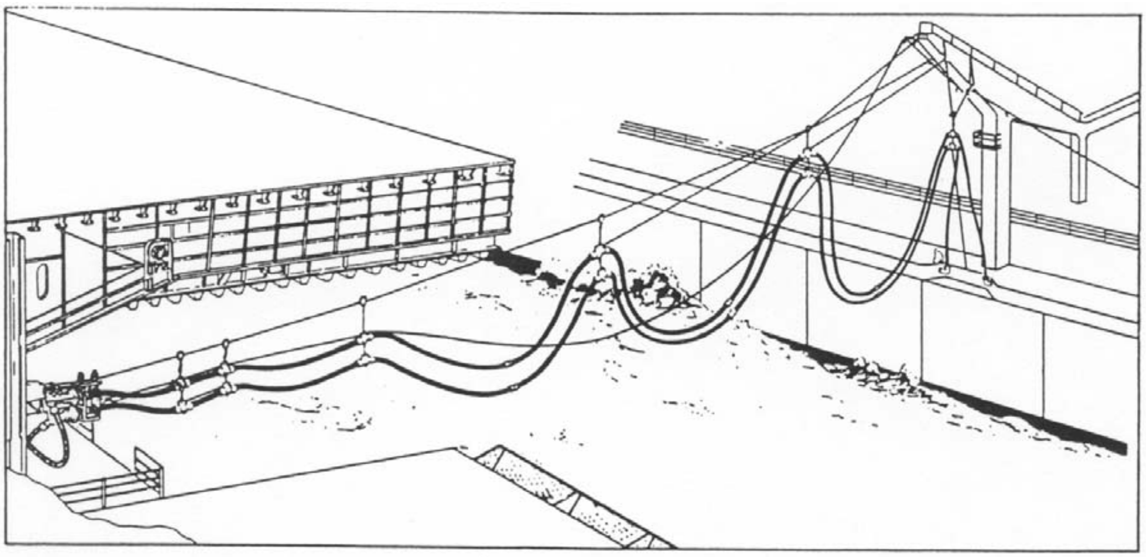


Figure 6.32 – STREAM hose rig

Current FAS procedures and hardware can be applied and are sufficient for use in skin-to-skin operations. The limiting factor for liquid cargo transfer in skin-to-skin operations is the ability to safely move the ships within close proximity of each other, not the passing of hoses for the transfer of fuel. Once the involved ships have reached the determined skin-to-skin standoff, which shall be much less than current UNREP standoff, FAS operations could then commence following existing procedures.

However, in a skin-to-skin connected replenishment operation, the close-in hose rig is a better option for fuel transfers because of the reduced standoff distance. The close-in configuration reduces the amount of hose and support hardware required to transfer fuel and is depicted in the following figure.

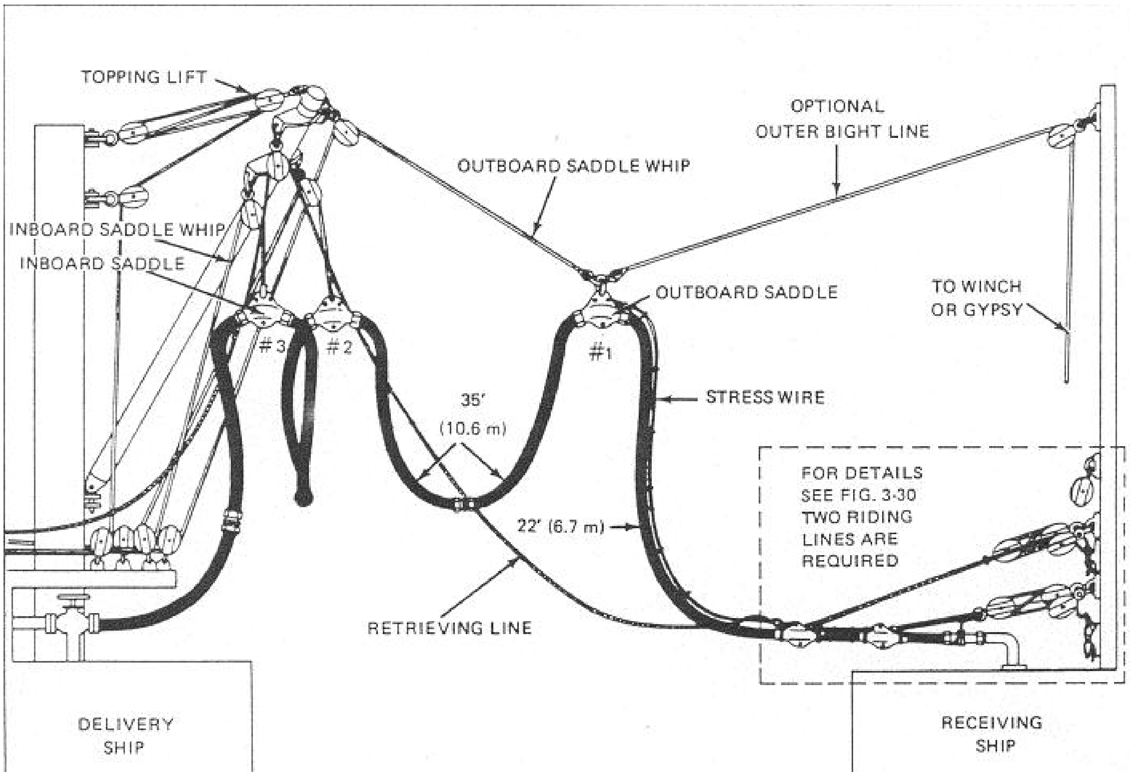


Figure 6.33 – Close-in hose rig

Another option would be the use of a dedicated hose handling cranes or davits. These cranes are specifically designed to be explosion proof and have precise step control for better manipulation. The standoff distances encountered in skin-to-skin operations would be an ideal application of these commercially available cranes.

Finally, it is important to note that skin-to-skin fuel transfers are regularly conducted in the shipping industry. Large oil tankers unload fuel to smaller tankers for transit into shallow rivers and canals. These transfers are performed underway at speeds of approximately 3-4 knots. These low speeds are possible because the ships involved are purpose built for the operation, and are equipped with thrusters that allow the ships to maintain control. The mooring and fendering systems are conventional as well. Standard

"Sea Cushion" fenders are used, mounted on fixed sliders on the sides of the ships to allow vertical movement. Controllable tension mooring winches are used to maintain the lines. Lastly, an experienced "mooring master" is appointed and given responsibility for the operation.

7. Concept Assessment and Integration

7.1. Feasibility Assessment

This section will assess the concepts presented in this study, discussing which are feasible, which are not, and how the different concepts would work alone or in combination with others to enable skin-to-skin operations. Table 7.1 provides a summary list of the technologies discussed in the report.

Table 7.1 – Technology Summary

Ship Control	Mooring and Fendering	Cargo Transfer
Rudder Roll Stabilization	Suspended Pneumatic Fenders	Pedestal Cranes – RBTS
Active Flume Tanks	Suspended Composite Box Fender	Pedestal Crane – PCS
Passive Flume Tanks	Suspended Extension Aarm	Self Leveling Pedestal Crane – PCS
Active Fins	Suspended Articulated Arm	Active Target Tracking
Dynamic Positioning	Suspended Parallel Fender	Gantry Cranes
Free Space Optical Communications & Ranging System	Surface Towed Fendering Sled	Bridges/Conveyors/ILP
Automated/Assisted Approach	Surface Towed SWATH Fender	Liquid
	Submerged Towed Fenders	Knuckle and Telescoping Boom Cranes
	Waterjet Repulsion	
	Static Mixer Repulsion	
	Vacuum Mooring	
	Advanced Mooring Winches/Conventional Mooring Line	
	Elastomeric Mooring Line	

Several of the concepts presented in this report have been determined to be not feasible, based on either technology limitations or practical concerns regarding the broad requirements for skin-to-skin cargo transfer. They are:

- Rudder roll stabilization

- Waterjet repulsion
- Static mixer repulsion
- Pedestal cranes w/RBTS
- Cargo Bridges/conveyors/ILP

Rudder roll stabilization is a proven, effective technology for reducing ship motions. However, its limitations make it a poor choice for application in a skin-to-skin operation. The effectiveness greatly depends on a variety of ship characteristics and operating conditions. Also, the use of the ship's steering mechanism for roll reduction during situations in which an obstacle, in this case another ship, is in close proximity is not advisable. The benefits of RRS are available using other systems, which do not pose the same problems.

Both waterjet repulsion and static mixer fendering systems have been deemed not feasible. The evaluation of the waterjet repulsion concept quickly concludes that this concept is feasible for ships close together at low speeds, but is not feasible for a lateral separation distance of 20 feet or speeds of 8 knots and above. The shearing vectors the waterjets would encounter at speeds of 8 knots and above would render the waterjets ineffective. The energy and pumping system, together with the large flow volume required, make this concept impractical. The evaluation of the static mixer repulsion concept also concludes that it is not feasible for the combination of speed and lateral distance required.

For cargo transfer, pedestal cranes using only the RBTS load stabilizing system will not be effective. Significant roll motions predicted by the modeling are normally approximately 1°, where an RBTS system is shown to be effective. However, in cases were the seas are approaching the ships at an angle, or where swell conditions separate from sea state are encountered, roll magnitudes can be expected to exceed 2°, with peak roll angles in extreme cases going much higher. Roll motions in this higher range drastically reduce the RBTS' effectiveness. In order to ensure a capable cargo transfer system in all likely conditions, cranes equipped only with the RBTS should not be considered.

The other cargo systems deemed not feasible are bridges, conveyors, and the ILP. The technical and operational problems posed by these systems appear too great to recommend developing them. The only feasible application of this concept would be a small pallet conveyor system with a specially designed receiving system that would allow the conveyor to avoid a direct interface with the receiving ship. The benefits and drawbacks of such a system would have to be studied to determine if this concept is worth pursuing.

Several of the concepts studied are feasible for virtually any ship application or ship-to-ship combination considered, and could provide benefits in a skin-to-skin operation. These include:

- Free Space Optical Communications & Ranging System
- Advanced mooring winches & conventional mooring lines
- Elastomeric mooring line
- Liquid cargo transfer
- Automated/assisted approach
- Knuckle and telescopic boom cranes

A succesful development program for the Free Space Optical Communications & Ranging System would enable a great deal of information to be monitored and shared between the ships involved in the operation. It is a low-impact system, requiring little in the way of ship modifications. The system would provide a great increase in capability from the current method of communications and ranging for UNREP, using the Phone and Distance (P&D) Line. The distance measure methods of a crewmember counting flags spaced 20' apart on the P&D line, or using commercially available handheld rangefinders, would be replaced with an automated, multiple position monitoring system that would provide real time distance and ship orientation data to the crew. The system could also be used to transfer ship operating data, such as forward speeds, GPS data, control surface information, and other potentially relevant data, perhaps for use in an automated approach and mooring system.

"Advanced mooring winches and conventional mooring line" simply refers to the practice of specifying winches and lines for a ship based on the requirements of skin-to-skin mooring, as opposed to the less demanding pierside or at anchor mooring. Skin-to-skin operations will require great line pull, speed, and response characteristics for the winches. Upon determining the requirements for a mooring system, based on the ship combinations and desired operating conditions, the proper equipment can be chosen. In addition to equipment design and specification, the requirement that the mooring system be robust enough to handle a variety of ship types and sizes needs to be determined. If, for instance, the decision is made that the MPF(F) must be able to conduct operations with ships ranging in size from other MPF(F) down to combatants, the mooring system must be adaptable to both situations, and all in between, perhaps by employing the methods discussed earlier.

Elastomeric mooring line presents an option for possibly reducing or eliminating the need for high speed mooring winches. The stress/strain properties of the proposed "SuperStretch" line would allow it to provide significant holding force while withstanding the relative motions of the ships. Specific mooring arrangements would have to be developed to determine the extent of its usefulness, but the potential is clear and warrants consideration.

Current liquid cargo transfer systems exist that will be effective during skin to skin operations. Specifically, the "Close-in hose rig" or hose handling cranes or davits would enable ships moored skin-to-skin to connect a hose for liquid transfer. One option not discussed previously is the potential for developing a higher transfer rate system. The short distances involved in skin-to-skin transfer would potentially allow for the use of a larger diameter hose, decreasing fueling times. This concept requires further investigation to determine the challenges and benefits.

An automated/assisted approach system is a desirable and feasible capability for installation on virtually any ship involved in skin-to-skin operations. Many ships already have integrated ship controls systems, as discussed earlier. The integration of an automated or assisted control system for skin-to-skin approach would greatly increase the safety and efficiency of the operation. A complete system would share information between the ships, using a system like the FSOCRS, and coordinate their controls.

The knuckle and telescoping boom cranes discussed in this report have several attractive features. Primarily is their small size and corresponding minimal ship arrangements impact. They can handle small loads in excess of the current UNREP equipment, and for the short distances envisioned for skin-to-skin operations have adequate reach. These cranes should be considered for installation on any ships, current or future, where they could contribute during a conceived skin-to-skin operation.

The remaining concepts are feasible for certain ships or ship combinations. They are:

- Dynamic positioning
- Flume tanks
- Active fins
- Fenders
- Vacuum pads
- Pedestal crane/PCS
- Self leveling pedestal crane
- Gantry cranes
- Active target tracking

Using a dynamic positioning system is a feasible option to help maintain specified standoff distances in a skin-to-skin operation. DP could also reduce the amount of required fendering and mooring if properly applied. In order for the system to be effective, a ship must be equipped with sufficient control systems and thrusters. DP system thrusters, because of their size and complexity, cannot be retrofitted on an existing ship and must be defined during the design phase of the ship. An integrated control system, along with new feed-forward sensors, would also need to be developed to accurately predict motion and maintain stand off.

Ship roll stabilization systems are effective technologies that would be of great benefit to skin-to-skin operations. Reducing the roll magnitude decreases the requirements on cargo handling, mooring, and fendering systems. Considering the results of the hydrodynamic study, situations where a ship may be rolling at 2°, a roll stabilization system operating at 50% effectiveness could cut peak rolls down to 1°. Some of the extreme cases where roll approached 4° would be reduced to 2°, moving from a marginal situation to a much better one. As shown in the pedestal crane pendulation section, a reduction in roll of this magnitude would provide a significant benefit in terms of crane working area and machinery requirements. Passive and active flume tanks and active fin stabilizers each have design features that make them better suited for different circumstances, but with proper planning most ships involved in skin-to-skin operations will be candidates for at least one of these systems.

Fendering systems are crucial for a successful skin-to-skin operation. Of those considered, the most promising are the SWATH towed fender and the suspended pneumatic fenders. The SWATH fender could provide adequate separation without placing heavy loads on the side of the ship that deploys them. The main complication of this concept is the large size that is likely. Suspended pneumatic fenders have the advantage of being commercially available, widely used products.

Vacuum mooring is a promising technology. Assuming the system could be designed and built to handle the loads and motions in a skin-to-skin operation, it would eliminate the need for time consuming line passing and rigging, as well as reducing the requirements on the fendering system. The type of system that would be required for skin-to-skin mooring lends itself to inclusion on a new-build ship, due to the large amount of machinery that must be installed.

Standard pedestal mounted boom cranes with the Pendulation Control System installed are the best, most robust option for skin-to-skin cargo transfer. Within the range of motions expected, PCS cranes will be capable of operating safely and efficiently. An advantage of the PCS system is that the cranes do not require significant modifications. Many currently installed cranes will likely require machinery upgrades to meet the required performance levels, but structural changes to the crane or the ship would most likely not be necessary. The self leveling pedestal crane with PCS is a concept that would provide excellent crane performance in very high ship motion conditions. However, the engineering challenges in designing and building such a system are great.

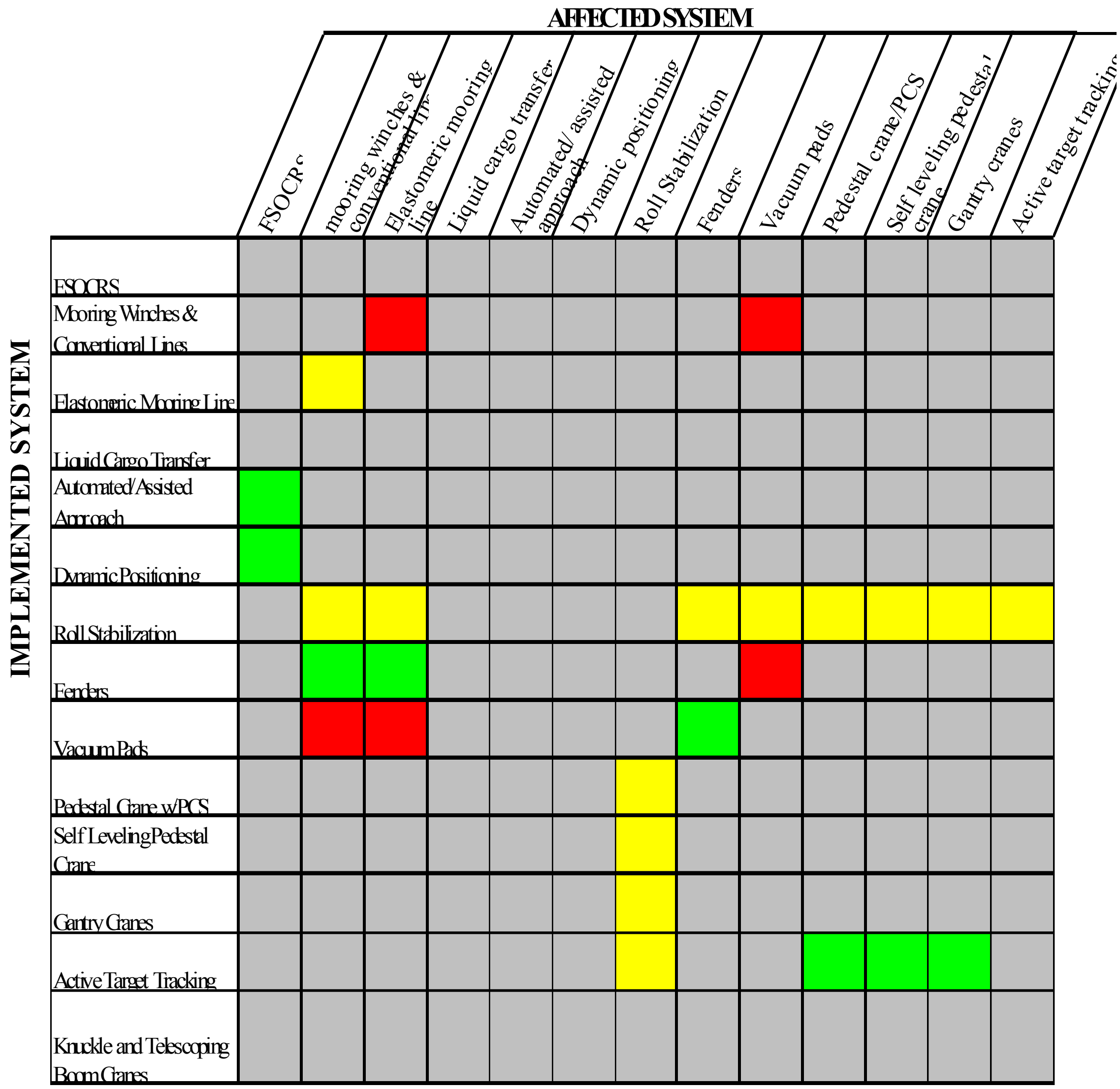
Ship mounted gantry cranes, designed with sufficient outboard reach, are another option for skin-to-skin cargo transfer. By their nature gantry cranes allow rapid movement of cargo in a controlled and safe manner. The drawbacks to gantry cranes are in the ship arrangements impact they present. They are quite large and heavy, and require tracks or rails on which to move. The only operational issue is that they would extend over the side of the ship, presenting a potential collision hazard with the other ship's structure. Effective approach and mooring procedures could minimize this risk.

Active target tracking is an intriguing technology. If successfully developed, it would greatly increase the ability of the crane operator to precisely place loads on the target ship. This could be especially useful when a large amount of cargo is being placed into a small area, or for situations such as VLS rearming where the target area is very small.

7.2. Concept Integration

Table 7.2 shows a matrix of the feasible technologies and their effects on the other technologies. Red blocks indicate a system that is rendered unnecessary by the implementation of the other. Yellow indicates a system that's required performance is increased, reduced, or otherwise altered by the other. Green means that the affected system (or one of multiple systems) is required in order to implement the other, and gray means the implementation of one system does not affect the other.

Table 7.2 – Technology Effects Matrix



The table shows how roll stabilization systems will have the most impact on the skin-to-skin operation. Implementing one of these systems will decrease the demands on any mooring winches, elastomeric lines, fenders, vacuum moorings, and all types of cranes involved. It also provides a margin of safety that otherwise will not be present. Note from the hydrodynamic models that while significant roll motions averaged in the 1°-2° range, certain cases see rolling as much as double that magnitude, and individual peak rolls would certainly be higher. Also note that this study primarily focused on sea state conditions, while long period swells are capable of exciting much greater roll, as shown in the limited models performed in this study. Roll stabilization has the capability to greatly reduce the magnitudes of roll experienced by each ship.

The mooring and fendering systems each affect the need for the others. The development of adequate mooring winches using conventional mooring line will eliminate the need for developing elastomeric lines or vacuum mooring, and require the use of a fendering system. Likewise, elastomeric mooring lines would reduce and perhaps eliminate the need for mooring winches or vacuum mooring. A vacuum mooring system will eliminate the need for either type of line mooring system, but would still require fendering. Each of these three approaches will need to be addressed when deciding what solution is best for a given skin-to-skin operation. Only a thorough study of the capabilities, ship impacts, and costs can determine which combination of systems to use.

8. Conclusions and Recommendations

Skin-to-skin connected replenishment in sea state 5 is feasible, if the proper combination of the concepts presented in this paper is implemented. These operations can only be accomplished with a "system of systems" approach. Mooring and fendering, cargo transfer, and ship control technologies must be selectively implemented to form a comprehensive system. In addition, the nature of each system must be considered when deciding how best to implement it. Some systems are not candidates for backfit onto existing ships, while others are equally suited to backfit or new build. This section will make recommendations for each technology and specify what aspects still require basic research, and which are advanced enough to pursue more directly. A summary of the main recommendations is provided in Table 8.1.

Several of the concepts presented in this paper are not feasible for skin-to-skin cargo operations, based on the requirements set forth and their technical and practical limitations, and do not warrant further consideration. They are: rudder roll stabilization, waterjet repulsion, static mixer repulsion, pedestal cranes w/RBTS, and cargo bridges/conveyors/ILP. Several of the fendering concepts discussed, the composite suspended fender, articulating fenders, and towed surface and submerged fenders, aren't entirely infeasible. However, some of the other systems discussed appear to be better solutions.

The Free Space Optical Communications & Ranging System, or another system providing like capabilities, should be developed and installed on all ships to be involved in skin-to-skin operations, as should an automated/assisted approach system. The process of bringing two ships together in SS5 is very dangerous. Attempting to do this with human observers reporting information through a chain of people who pass commands along to a helmsman, on two different ships, is simply too slow and prone to mistakes. Automatically monitoring information, sharing it between ships, and presenting it in a useable format is the minimum recommended system. The development of an automated approach system is also recommended, however only in conjunction with the assisted system. Most of the equipment necessary for implementing these systems is currently available. As discussed earlier, further research is needed to complete the FSOCRS design. Once that is accomplished, the challenge will be in choosing the proper components, and integrating them into a useful system.

A dynamic positioning system is a feasible option, if the proper planning and integration is performed early in the design phase of a new ship. Ships such as the MPF (F) should be equipped with the required machinery and systems for effective DP. Such a system could substantially reduce the demands on skin-to-skin mooring and fendering systems, and enhance the safety of the approach section as well. Dynamic Positioning can be implemented with currently available technology.

The mooring and fendering system for a skin-to-skin operation must be developed with individual ship-to-ship combinations in mind. Once these combinations are specified, each can be studied to determine which mooring and fendering system is most appropriate. When the desired method is selected, the design of the system can be performed. Each of the technologies presented in this paper have the capacity to meet the needs of a skin-to-skin operation. Winches and conventional line are the lowest risk systems, due to the fully developed nature of those products. The SWATH towed fender and suspended pneumatic fenders are also feasible. Development of a SWATH fender would require a significant effort to study the necessary shapes, construction, and hydrodynamic characteristics before a system suitable for testing would be feasible. A system composed of suspended pneumatic fenders must be more thoroughly evaluated to determine if it would be sufficient. Elastomeric line and vacuum mooring have proven effective in other applications but require further development. The recommended course of action is to pursue the suspended pneumatic fendering system in combination with conventional mooring winches and lines. This represents the lowest risk option and does not require an investment in basic research. If it is determined that such a system would not be suitable, the SWATH towed fender, vacuum mooring, and elastomeric lines should be reevaluated.

Roll stabilization should be studied and implemented whenever possible on ships involved in skin-to-skin operations. The benefits of these systems are clear, and if installed during construction the costs are relatively modest. The passive tank system in particular is attractive because it works at low ship speeds, does not require much maintenance or power, is inexpensive to build, and is contained within the ship. This is a well developed technology that only requires proper planning to successfully implement.

The results shown previously detail what level of crane performance is required for PCS operation in different ship motion conditions. A comprehensive study of the currently deployed cranes will be necessary to determine what capability each possesses, and what performance upgrades, if any, are required. Cranes to be installed on future ships can be designed for the necessary performance prior to installation. The recommended approach is to continue development of the PCS and study the requirements for it's installation aboard the cranes of any sea base ship. While the self leveling PCS crane would be capable of performing in large ship motion conditions, the engineering challenges and costs appear to outweigh the benefits, especially considering the magnitudes of motion predicted in this study. Active target tracking is another technology that would clearly provide a benefit, however it has not yet been determined if this capability is necessary. Further refinement of requirements should be completed before proceeding with development of this system.

Gantry cranes should be considered for any future construction ship on which they would be appropriate. For certain applications, such as servicing one or more container holds on a ships such as the MPS Waterman Class and LASH, they may be the ideal choice. The challenge with these cranes is to build them with adequate outreach for a skin-to-skin operation while not making them so heavy as to drastically impact the ship. Further investigation should be performed to determine what capabilities the manufacturers are capable of providing, above and beyond those of their current models.

Knuckle and telescoping boom cranes have the capability to provide a robust cargo transfer capability on a wide variety of ships. Their ability to handle moderate sized loads at outreaches up to 20m makes them an ideal candidate for transferring stores, munitions, and other breakbulk cargo during a skin-to-skin operation. The implementation of these cranes should be considered for any seabase ships that could make use of them. They are available as COTS items and would require minimal effort to design and install.

The final recommendation is that the requirements for skin-to-skin operations be fully developed before implementing any technologies or procedures on a large scale. Each and every technology discussed in this paper has capabilities and limitations that can only be fully addressed if the scenario they will be operating in is well defined. Additionally, a body of knowledge needs to be developed before making decisions about which of these technologies would be suitable for applications which they were not originally intended. This can be achieved by developing smaller, limited requirements for testing of specific technologies. For instance, experiments such as choosing a set of two cargo ships, such as a T-ACS and LMSR, and developing a fendering arrangement to test the applicability of pneumatic fenders, mooring winches and lines, and performing a demonstration of skin-to-skin cargo transfer using PCS equipped boom cranes and knuckle boom cranes would provide valuable experience and knowledge that does not currently exist. A similar experiment could be conducted to determine the usefulness of the STREAM system and small cranes if a combatant ship is included.

Table 8.1 - Recommendations

Subject	Recommendation
Free Space Optical Communications & Ranging System	Continued research towards fully functional and deployable system.
Automated/Assisted Approach	Perform detailed investigation into available technologies and integration issues.
Dynamic Positioning	Install the equipment required for this capability on new-build ships which will be involved in skin-to-skin operations.
Conventional Mooring System using Pneumatic Fendering System, Mooring Winches, and Lines	Perform detailed mooring and fendering analysis for a small number of specific examples. Determine if this arrangement is feasible.
SWATH Fender	Perform necessary research and development of this concept if the pneumatic fendering system proves to be unsuccessful or not completely sufficient.
Vacuum Mooring	Perform necessary research and development of this concept if the conventional system proves to be unsuccessful or not completely sufficient.
Elastomeric Mooring Line	Perform necessary research and development of this concept if the conventional system proves to be unsuccessful or not completely sufficient.
Roll Stabilization	Install the equipment required for this capability on new-build ships which will be involved in skin-to-skin operations, and investigate the possibility of back-fitting a system onto existing ships where feasible.
PCS Pedestal Boom Crane	Install PCS capability on all pedestal boom cranes on board ships that will be involved in skin-to-skin operations.
Gantry Cranes	Investigate the ability of gantry cranes to meet the requirements for skin-to-skin operation where they exist on current ships, and consider their installation on new-build ships which will be involved in skin-to-skin operations.
Knuckle and Telescoping Boom Cranes	Investigate the ability of these cranes to meet the requirements for skin-to-skin operation. Where applicable, install on existing and new-build ships which will be involved in skin-to-skin operations.
Experiments	Design focused experiments with the goal of testing a limited number of enabling technologies.
Requirements	Continue development of a detailed skin-to-skin concept of operations. This will help define the requirements, which must be clarified before committing significant resources to any proposed solutions.

[1] Frank H. Sellars & John P. Martin, Selection and Evaluation of Ship Roll Stabilization Systems; February 14, 1991, 12

[2] Sellars & Martin, 9-13
[3] Sellars & Martin, 17
[4] Sellars & Martin, 17
[5] Sellars & Martin, 17
[6] Intelligent Systems Division, National Institutes of Standards and Technology (NIST), RoboCrane® for Portable Ports and Mobile Offshore Base; January 17, 1997

Feasibility Study for Sea State 5 Skin-To-Skin Cargo Transfer Operations

APPENDICES A - E

Submitted by:

Naval Surface Warfare Center – Carderock Division
Code 2820 Expeditionary/Logistics Systems

In conjunction with:

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APPENDIX B – T-ACS 5 AND LMSR UNCONNECTED

WAMIT irregular seas response calculations: WAMIRGW
WAMIRGW input file: TACWAT0.IRG
WAMIT .4 file: TACWATY0.4
number of headings = 8 number of periods = 32 2
Reference length = 581.8300 Units conv 1.0000
Output file: TACWATY0.OVT
Input units: FEET Output units: FEET Angles in degrees (deg/sec or deg/sec^2)

SEA STATE 3
Bretshnieder seas> H1/3= 4.100 To = 8.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:
WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.074	0.099	0.288	0.050	0.143	0.028
135.000	0.077	0.098	0.276	0.029	0.121	0.042
90.000	0.016	0.305	0.608	0.083	0.097	0.043
45.000	0.095	0.091	0.188	0.051	0.129	0.044
0.000	0.094	0.091	0.190	0.042	0.156	0.039
315.000	0.180	0.115	0.307	0.083	0.294	0.055
270.000	0.042	0.464	1.368	0.101	0.094	0.042
225.000	0.141	0.121	0.434	0.091	0.294	0.065

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:
WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.024	0.041	0.123	0.026	0.039	0.005
135.000	0.046	0.063	0.198	0.089	0.058	0.023
90.000	0.014	0.404	0.869	0.033	0.015	0.018
45.000	0.051	0.042	0.166	0.084	0.060	0.019
0.000	0.039	0.042	0.113	0.029	0.047	0.008
315.000	0.053	0.061	0.141	0.089	0.064	0.021
270.000	0.013	0.348	0.754	0.045	0.017	0.015
225.000	0.048	0.077	0.174	0.077	0.067	0.021

Absolute Point 1 TACS connect #1
TACS xyz 195.500 -39.000 20.420 (ref GDF origin)
LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.060	0.141	0.754
135.000	0.055	0.205	0.650
90.000	0.029	0.373	0.713
45.000	0.067	0.102	0.374
0.000	0.065	0.087	0.404
315.000	0.106	0.190	1.147
270.000	0.067	0.543	1.512
225.000	0.102	0.274	1.271

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.015	0.042	0.242
135.000	0.035	0.072	0.352
90.000	0.028	0.424	0.900
45.000	0.042	0.064	0.208
0.000	0.027	0.022	0.111
315.000	0.036	0.085	0.266
270.000	0.013	0.371	0.777
225.000	0.040	0.081	0.348

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.060	0.107	0.210
135.000	0.055	0.120	0.201
90.000	0.029	0.345	0.673
45.000	0.067	0.204	0.484
0.000	0.065	0.185	0.580
315.000	0.106	0.219	0.718
270.000	0.067	0.472	1.272
225.000	0.102	0.188	0.680

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.015	0.040	0.084
135.000	0.035	0.085	0.239
90.000	0.028	0.388	0.850
45.000	0.042	0.064	0.228
0.000	0.027	0.053	0.223
315.000	0.036	0.075	0.305
270.000	0.013	0.338	0.752
225.000	0.040	0.098	0.193

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.112	0.130	0.754
135.000	0.065	0.208	0.650
90.000	0.093	0.423	0.713
45.000	0.074	0.088	0.374
0.000	0.101	0.086	0.404
315.000	0.177	0.165	1.147
270.000	0.126	0.594	1.512
225.000	0.242	0.267	1.271

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.024	0.044	0.242
135.000	0.047	0.085	0.352
90.000	0.032	0.420	0.900
45.000	0.028	0.095	0.208
0.000	0.018	0.025	0.111
315.000	0.040	0.114	0.266
270.000	0.019	0.377	0.777
225.000	0.036	0.087	0.348

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.112	0.105	0.210
135.000	0.065	0.134	0.201
90.000	0.093	0.398	0.673
45.000	0.074	0.236	0.484
0.000	0.101	0.198	0.580
315.000	0.177	0.267	0.718
270.000	0.126	0.521	1.272
225.000	0.242	0.216	0.680

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.024	0.037	0.084
135.000	0.047	0.091	0.239
90.000	0.032	0.383	0.850
45.000	0.028	0.078	0.228
0.000	0.018	0.046	0.223
315.000	0.040	0.081	0.305
270.000	0.019	0.342	0.752
225.000	0.036	0.105	0.193

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)

LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.064	0.159	0.881	0.884	0.068	0.031
135.000	0.069	0.213	0.927	0.926	0.105	0.041
90.000	0.040	0.614	1.340	1.301	0.087	0.058
45.000	0.087	0.075	0.412	0.415	0.127	0.035
0.000	0.081	0.099	0.486	0.483	0.051	0.040
315.000	0.120	0.207	1.327	1.323	0.087	0.055
270.000	0.068	0.683	2.083	2.106	0.126	0.050
225.000	0.110	0.254	1.142	1.159	0.124	0.063

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)

LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.064	0.128	0.271	0.296	0.068	0.031
135.000	0.069	0.105	0.319	0.323	0.105	0.041
90.000	0.040	0.531	1.181	1.169	0.087	0.058
45.000	0.087	0.184	0.667	0.648	0.127	0.035
0.000	0.081	0.207	0.703	0.713	0.051	0.040
315.000	0.120	0.222	0.502	0.520	0.087	0.055
270.000	0.068	0.604	1.877	1.902	0.126	0.050
225.000	0.110	0.215	0.818	0.834	0.124	0.063

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)

LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.127	0.146	0.881	0.891	0.068	0.031
135.000	0.098	0.237	0.927	0.916	0.105	0.041
90.000	0.103	0.648	1.340	1.302	0.087	0.058
45.000	0.090	0.103	0.412	0.427	0.127	0.035
0.000	0.114	0.100	0.486	0.497	0.051	0.040
315.000	0.173	0.197	1.327	1.330	0.087	0.055
270.000	0.130	0.735	2.083	2.116	0.126	0.050
225.000	0.238	0.231	1.142	1.173	0.124	0.063

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)

LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.127	0.115	0.271	0.294	0.068	0.031
135.000	0.098	0.161	0.319	0.313	0.105	0.041
90.000	0.103	0.566	1.181	1.174	0.087	0.058
45.000	0.090	0.268	0.667	0.621	0.127	0.035
0.000	0.114	0.205	0.703	0.715	0.051	0.040
315.000	0.173	0.232	0.502	0.523	0.087	0.055
270.000	0.130	0.643	1.877	1.906	0.126	0.050
225.000	0.238	0.230	0.818	0.840	0.124	0.063

SEA STATE 4

Bretshnieder seas> H1/3= 8.200 To = 9.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.269	0.216	0.682	0.097	0.363	0.059
135.000	0.274	0.283	0.701	0.099	0.321	0.116
90.000	0.028	0.835	1.531	0.231	0.167	0.078
45.000	0.319	0.254	0.494	0.157	0.338	0.127
0.000	0.310	0.199	0.448	0.089	0.380	0.086
315.000	0.502	0.259	0.861	0.172	0.700	0.127
270.000	0.080	1.074	2.867	0.264	0.169	0.077
225.000	0.415	0.272	1.123	0.194	0.695	0.155

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.087	0.076	0.285	0.052	0.098	0.010
135.000	0.180	0.181	0.459	0.209	0.163	0.070
90.000	0.029	0.969	1.973	0.065	0.028	0.031
45.000	0.186	0.141	0.386	0.197	0.163	0.065
0.000	0.117	0.074	0.261	0.060	0.112	0.014
315.000	0.194	0.164	0.333	0.203	0.171	0.068
270.000	0.028	0.895	1.811	0.088	0.030	0.027
225.000	0.182	0.191	0.396	0.171	0.175	0.064

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.204	0.295	1.855
135.000	0.208	0.566	1.651
90.000	0.052	1.014	1.743
45.000	0.237	0.330	1.067
0.000	0.230	0.211	1.059
315.000	0.322	0.457	2.778
270.000	0.124	1.262	3.140
225.000	0.292	0.622	3.079

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.062	0.077	0.587
135.000	0.136	0.215	0.863
90.000	0.053	1.014	2.018
45.000	0.162	0.216	0.615
0.000	0.086	0.038	0.269
315.000	0.140	0.247	0.663
270.000	0.030	0.942	1.847
225.000	0.155	0.221	0.872

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.204	0.253	0.560
135.000	0.208	0.366	0.598
90.000	0.052	0.886	1.574
45.000	0.237	0.576	1.232
0.000	0.230	0.406	1.363
315.000	0.322	0.490	1.752
270.000	0.124	1.102	2.694
225.000	0.292	0.483	1.670

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.062	0.075	0.205
135.000	0.136	0.271	0.611
90.000	0.053	0.943	1.932
45.000	0.162	0.230	0.544
0.000	0.086	0.096	0.528
315.000	0.140	0.245	0.746
270.000	0.030	0.878	1.800
225.000	0.155	0.276	0.451

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.235	0.284	1.855
135.000	0.151	0.580	1.651
90.000	0.161	1.164	1.743
45.000	0.183	0.295	1.067
0.000	0.218	0.209	1.059
315.000	0.376	0.420	2.778
270.000	0.230	1.417	3.140
225.000	0.529	0.607	3.079

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.059	0.086	0.587
135.000	0.117	0.252	0.863
90.000	0.061	1.019	2.018
45.000	0.090	0.275	0.615
0.000	0.047	0.051	0.269
315.000	0.111	0.311	0.663
270.000	0.039	0.962	1.847
225.000	0.098	0.235	0.872

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.235	0.263	0.560
135.000	0.151	0.425	0.598
90.000	0.161	1.040	1.574
45.000	0.183	0.680	1.232
0.000	0.218	0.443	1.363
315.000	0.376	0.593	1.752
270.000	0.230	1.254	2.694
225.000	0.529	0.571	1.670

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.059	0.072	0.205
135.000	0.117	0.258	0.611
90.000	0.061	0.946	1.932
45.000	0.090	0.234	0.544
0.000	0.047	0.088	0.528
315.000	0.111	0.236	0.746
270.000	0.039	0.894	1.800
225.000	0.098	0.276	0.451

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)

LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.191	0.311	2.058	2.063	0.129	0.064
135.000	0.249	0.545	2.184	2.165	0.262	0.103
90.000	0.069	1.431	2.982	2.820	0.234	0.103
45.000	0.287	0.181	0.962	0.957	0.326	0.089
0.000	0.238	0.227	1.186	1.184	0.100	0.086
315.000	0.354	0.458	3.150	3.141	0.203	0.120
270.000	0.127	1.462	4.410	4.395	0.297	0.090
225.000	0.328	0.531	2.651	2.667	0.286	0.136

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)

LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.191	0.287	0.685	0.744	0.129	0.064
135.000	0.249	0.240	0.712	0.729	0.262	0.103
90.000	0.069	1.222	2.653	2.539	0.234	0.103
45.000	0.287	0.464	1.558	1.513	0.326	0.089
0.000	0.238	0.428	1.580	1.592	0.100	0.086
315.000	0.354	0.442	1.184	1.211	0.203	0.120
270.000	0.127	1.292	4.033	4.029	0.297	0.090
225.000	0.328	0.449	1.924	1.951	0.286	0.136

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)

LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.262	0.291	2.058	2.074	0.129	0.064
135.000	0.225	0.629	2.184	2.124	0.262	0.103
90.000	0.176	1.535	2.982	2.822	0.234	0.103
45.000	0.228	0.238	0.962	0.984	0.326	0.089
0.000	0.234	0.230	1.186	1.207	0.100	0.086
315.000	0.376	0.483	3.150	3.144	0.203	0.120
270.000	0.240	1.598	4.410	4.411	0.297	0.090
225.000	0.525	0.471	2.651	2.706	0.286	0.136

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)

LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.262	0.275	0.685	0.740	0.129	0.064
135.000	0.225	0.372	0.712	0.709	0.262	0.103
90.000	0.176	1.326	2.653	2.548	0.234	0.103
45.000	0.228	0.680	1.558	1.424	0.326	0.089
0.000	0.234	0.432	1.580	1.586	0.100	0.086
315.000	0.376	0.466	1.184	1.206	0.203	0.120
270.000	0.240	1.408	4.033	4.035	0.297	0.090
225.000	0.525	0.536	1.924	1.943	0.286	0.136

SEA STATE 5

Bretshnieder seas> H1/3= 13.120 To = 10.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.661	0.367	1.231	0.157	0.680	0.096
135.000	0.652	0.617	1.341	0.265	0.612	0.232
90.000	0.038	1.650	2.783	0.502	0.229	0.112
45.000	0.728	0.555	1.011	0.363	0.645	0.259
0.000	0.726	0.342	0.849	0.155	0.697	0.143
315.000	0.994	0.530	1.707	0.327	1.195	0.237
270.000	0.117	1.958	4.528	0.552	0.238	0.109
225.000	0.861	0.541	2.094	0.383	1.185	0.287

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.263	0.109	0.501	0.080	0.200	0.014
135.000	0.460	0.407	0.824	0.351	0.336	0.153
90.000	0.045	1.778	3.350	0.117	0.039	0.043
45.000	0.470	0.355	0.712	0.332	0.332	0.146
0.000	0.301	0.105	0.451	0.094	0.218	0.020
315.000	0.485	0.371	0.643	0.335	0.345	0.152
270.000	0.044	1.692	3.179	0.145	0.043	0.038
225.000	0.462	0.407	0.736	0.282	0.350	0.143

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.514	0.482	3.384
135.000	0.516	1.152	3.036
90.000	0.075	1.989	3.117
45.000	0.563	0.760	2.241
0.000	0.566	0.381	2.084
315.000	0.693	0.926	4.794
270.000	0.177	2.305	4.920
225.000	0.628	1.123	5.388

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.206	0.112	1.107
135.000	0.359	0.500	1.629
90.000	0.078	1.857	3.399
45.000	0.417	0.511	1.302
0.000	0.236	0.052	0.581
315.000	0.370	0.539	1.306
270.000	0.048	1.774	3.223
225.000	0.402	0.499	1.688

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.514	0.459	1.173
135.000	0.516	0.821	1.324
90.000	0.075	1.741	2.804
45.000	0.563	1.176	2.299
0.000	0.566	0.680	2.433
315.000	0.693	0.927	3.122
270.000	0.177	2.035	4.290
225.000	0.628	1.006	2.962

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.206	0.109	0.419
135.000	0.359	0.622	1.177
90.000	0.078	1.751	3.288
45.000	0.417	0.560	1.030
0.000	0.236	0.137	0.950
315.000	0.370	0.580	1.404
270.000	0.048	1.675	3.157
225.000	0.402	0.610	0.857

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.412	0.486	3.384
135.000	0.304	1.208	3.036
90.000	0.221	2.318	3.117
45.000	0.368	0.725	2.241
0.000	0.400	0.388	2.084
315.000	0.616	0.899	4.794
270.000	0.327	2.648	4.920
225.000	0.860	1.124	5.388

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.142	0.130	1.107
135.000	0.255	0.541	1.629
90.000	0.089	1.889	3.399
45.000	0.238	0.571	1.302
0.000	0.134	0.078	0.581
315.000	0.257	0.619	1.306
270.000	0.061	1.823	3.223
225.000	0.240	0.502	1.688

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.412	0.500	1.173
135.000	0.304	0.989	1.324
90.000	0.221	2.077	2.804
45.000	0.368	1.415	2.299
0.000	0.400	0.755	2.433
315.000	0.616	1.132	3.122
270.000	0.327	2.377	4.290
225.000	0.860	1.224	2.962

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.142	0.106	0.419
135.000	0.255	0.588	1.177
90.000	0.089	1.783	3.288
45.000	0.238	0.555	1.030
0.000	0.134	0.131	0.950
315.000	0.257	0.558	1.404
270.000	0.061	1.722	3.157
225.000	0.240	0.603	0.857

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.404	0.486	3.505	3.512	0.193	0.103
135.000	0.577	1.027	3.705	3.649	0.483	0.192
90.000	0.094	2.461	4.903	4.537	0.483	0.145
45.000	0.634	0.352	1.727	1.683	0.613	0.165
0.000	0.485	0.400	2.134	2.133	0.157	0.140
315.000	0.731	0.835	5.304	5.283	0.376	0.207
270.000	0.183	2.438	6.948	6.842	0.572	0.128
225.000	0.692	0.856	4.440	4.430	0.541	0.225

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.404	0.505	1.294	1.400	0.193	0.103
135.000	0.577	0.435	1.236	1.278	0.483	0.192
90.000	0.094	2.116	4.411	4.120	0.483	0.145
45.000	0.634	0.852	2.670	2.599	0.613	0.165
0.000	0.485	0.697	2.640	2.659	0.157	0.140
315.000	0.731	0.707	2.065	2.091	0.376	0.207
270.000	0.183	2.159	6.424	6.344	0.572	0.128
225.000	0.692	0.768	3.277	3.312	0.541	0.225

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.415	0.470	3.505	3.524	0.193	0.103
135.000	0.428	1.222	3.705	3.553	0.483	0.192
90.000	0.241	2.678	4.903	4.548	0.483	0.145
45.000	0.461	0.434	1.727	1.740	0.613	0.165
0.000	0.382	0.411	2.134	2.163	0.157	0.140
315.000	0.636	0.942	5.304	5.265	0.376	0.207
270.000	0.343	2.721	6.948	6.871	0.572	0.128
225.000	0.873	0.766	4.440	4.507	0.541	0.225

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.415	0.512	1.294	1.397	0.193	0.103
135.000	0.428	0.665	1.236	1.259	0.483	0.192
90.000	0.241	2.331	4.411	4.140	0.483	0.145
45.000	0.461	1.254	2.670	2.414	0.613	0.165
0.000	0.382	0.720	2.640	2.640	0.157	0.140
315.000	0.636	0.772	2.065	2.075	0.376	0.207
270.000	0.343	2.415	6.424	6.356	0.572	0.128
225.000	0.873	1.008	3.277	3.266	0.541	0.225

APPENDIX C – T-ACS 5 AND LMSR USNS WATSON SWAY AND YAW RESTRAINT

WAMIT irregular seas response calculations: WAMIRGW
WAMIRGW input file: TACWAT1.IRG
WAMIT .4 file: TACWATY1.4T
number of headings = 5 number of periods = 32 2
Reference length = 581.8300 Units conv 1.0000
Output file: TACWATY1.OVT
Input units: FEET Output units: FEET Angles in degrees (deg/sec or deg/sec^2)

SEA STATE 3

Bretshnieder seas> H1/3= 4.100 To = 8.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:
WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.075	0.074	0.324	0.215	0.145	0.003
135.000	0.080	0.076	0.304	0.177	0.128	0.021
90.000	0.006	0.213	0.724	0.753	0.042	0.009
45.000	0.100	0.054	0.219	0.111	0.140	0.021
0.000	0.096	0.042	0.195	0.092	0.160	0.008

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:
WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.024	0.040	0.131	0.046	0.039	0.003
135.000	0.047	0.061	0.187	0.077	0.056	0.021
90.000	0.013	0.382	0.790	0.112	0.010	0.009
45.000	0.052	0.052	0.156	0.067	0.059	0.021
0.000	0.038	0.023	0.111	0.037	0.046	0.008

Absolute Point 1 TACS connect #1
TACS xyz 195.500 -39.000 20.420 (ref GDF origin)
LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.063	0.035	0.788
135.000	0.063	0.067	0.685
90.000	0.020	0.363	1.083
45.000	0.068	0.066	0.342
0.000	0.059	0.013	0.379

Absolute Point 2 LMSR connect #1
LMSR xyz 195.500 53.000 20.420 (ref GDF origin)
LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.016	0.035	0.245
135.000	0.036	0.067	0.332
90.000	0.020	0.363	0.883
45.000	0.045	0.066	0.212
0.000	0.029	0.013	0.119

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.063	0.034	0.186
135.000	0.063	0.081	0.197
90.000	0.020	0.340	0.965
45.000	0.068	0.076	0.574
0.000	0.059	0.045	0.621

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.016	0.034	0.086
135.000	0.036	0.081	0.232
90.000	0.020	0.340	0.852
45.000	0.045	0.076	0.218
0.000	0.029	0.045	0.219

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.127	0.160	0.788
135.000	0.089	0.128	0.685
90.000	0.048	0.839	1.083
45.000	0.075	0.097	0.342
0.000	0.100	0.062	0.379

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.026	0.044	0.245
135.000	0.042	0.076	0.332
90.000	0.024	0.303	0.883
45.000	0.028	0.087	0.212
0.000	0.019	0.032	0.119

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.127	0.163	0.186
135.000	0.089	0.152	0.197
90.000	0.048	0.815	0.965
45.000	0.075	0.109	0.574
0.000	0.100	0.076	0.621

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.026	0.041	0.086
135.000	0.042	0.075	0.232
90.000	0.024	0.282	0.852
45.000	0.028	0.063	0.218
0.000	0.019	0.057	0.219

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.064	0.000	0.885	0.887	0.258	0.000
135.000	0.071	0.000	0.897	0.900	0.175	0.000
90.000	0.036	0.000	1.335	1.335	0.860	0.000
45.000	0.082	0.000	0.403	0.406	0.095	0.000
0.000	0.073	0.000	0.467	0.468	0.125	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.064	0.000	0.254	0.259	0.258	0.000
135.000	0.071	0.000	0.340	0.344	0.175	0.000
90.000	0.036	0.000	1.098	1.097	0.860	0.000
45.000	0.082	0.000	0.728	0.731	0.095	0.000
0.000	0.073	0.000	0.723	0.727	0.125	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.135	0.178	0.885	0.903	0.258	0.000
135.000	0.115	0.121	0.897	0.905	0.175	0.000
90.000	0.065	0.594	1.335	1.277	0.860	0.000
45.000	0.091	0.066	0.403	0.409	0.095	0.000
0.000	0.110	0.086	0.467	0.483	0.125	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.135	0.178	0.254	0.300	0.258	0.000
135.000	0.115	0.121	0.340	0.349	0.175	0.000
90.000	0.065	0.594	1.098	1.019	0.860	0.000
45.000	0.091	0.066	0.728	0.733	0.095	0.000
0.000	0.110	0.087	0.723	0.729	0.125	0.000

SEA STATE 4

Bretshnieder seas> H1/3= 8.200 To = 9.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.267	0.157	0.736	0.412	0.364	0.007
135.000	0.282	0.229	0.752	0.409	0.333	0.068
90.000	0.012	0.619	1.771	1.798	0.084	0.017
45.000	0.330	0.165	0.551	0.275	0.360	0.070
0.000	0.313	0.103	0.466	0.226	0.390	0.018

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.087	0.076	0.295	0.093	0.098	0.007
135.000	0.184	0.180	0.434	0.181	0.163	0.068
90.000	0.027	0.928	1.816	0.242	0.017	0.017
45.000	0.189	0.161	0.365	0.156	0.163	0.070
0.000	0.115	0.048	0.257	0.084	0.109	0.018

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.196	0.075	1.927
135.000	0.229	0.216	1.776
90.000	0.039	0.898	2.737
45.000	0.245	0.224	0.969
0.000	0.211	0.029	0.956

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.064	0.075	0.597
135.000	0.144	0.216	0.839
90.000	0.039	0.897	2.009
45.000	0.171	0.224	0.618
0.000	0.089	0.029	0.278

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.196	0.067	0.464
135.000	0.229	0.269	0.506
90.000	0.039	0.847	2.472
45.000	0.245	0.261	1.445
0.000	0.211	0.097	1.481

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.064	0.067	0.217
135.000	0.144	0.269	0.597
90.000	0.039	0.847	1.960
45.000	0.171	0.261	0.533
0.000	0.089	0.097	0.528

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.263	0.294	1.927
135.000	0.196	0.320	1.776
90.000	0.096	1.979	2.737
45.000	0.176	0.237	0.969
0.000	0.209	0.147	0.956

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.061	0.100	0.597
135.000	0.111	0.237	0.839
90.000	0.045	0.794	2.009
45.000	0.093	0.268	0.618
0.000	0.049	0.073	0.278

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.263	0.301	0.464
135.000	0.196	0.360	0.506
90.000	0.096	1.932	2.472
45.000	0.176	0.352	1.445
0.000	0.209	0.180	1.481

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.061	0.091	0.217
135.000	0.111	0.248	0.597
90.000	0.045	0.748	1.960
45.000	0.093	0.224	0.533
0.000	0.049	0.129	0.528

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.181	0.000	2.063	2.067	0.499	0.000
135.000	0.245	0.000	2.127	2.139	0.392	0.000
90.000	0.068	0.000	3.016	3.016	2.028	0.000
45.000	0.272	0.000	0.911	0.920	0.242	0.000
0.000	0.214	0.000	1.100	1.103	0.301	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.181	0.000	0.607	0.624	0.499	0.000
135.000	0.245	0.000	0.723	0.748	0.392	0.000
90.000	0.068	0.000	2.545	2.544	2.028	0.000
45.000	0.272	0.000	1.685	1.704	0.242	0.000
0.000	0.214	0.000	1.634	1.647	0.301	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.278	0.345	2.063	2.086	0.499	0.000
135.000	0.255	0.271	2.127	2.138	0.392	0.000
90.000	0.124	1.401	3.016	2.861	2.028	0.000
45.000	0.218	0.167	0.911	0.914	0.242	0.000
0.000	0.223	0.208	1.100	1.128	0.301	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.278	0.345	0.607	0.670	0.499	0.000
135.000	0.255	0.270	0.723	0.739	0.392	0.000
90.000	0.124	1.401	2.545	2.346	2.028	0.000
45.000	0.218	0.167	1.685	1.700	0.242	0.000
0.000	0.223	0.208	1.634	1.640	0.301	0.000

SEA STATE 5

Bretshnieder seas> H1/3= 13.120 To = 10.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.657	0.259	1.297	0.635	0.679	0.011
135.000	0.666	0.522	1.411	0.709	0.630	0.151
90.000	0.017	1.292	3.141	3.153	0.127	0.025
45.000	0.745	0.398	1.083	0.540	0.675	0.156
0.000	0.729	0.186	0.875	0.419	0.710	0.029

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.262	0.110	0.513	0.147	0.201	0.011
135.000	0.470	0.412	0.793	0.306	0.340	0.151
90.000	0.043	1.716	3.131	0.396	0.024	0.025
45.000	0.477	0.386	0.689	0.261	0.335	0.156
0.000	0.300	0.073	0.446	0.141	0.214	0.029

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.492	0.120	3.483
135.000	0.561	0.513	3.342
90.000	0.058	1.689	4.951
45.000	0.587	0.532	2.029
0.000	0.532	0.049	1.858

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.206	0.120	1.132
135.000	0.378	0.514	1.630
90.000	0.058	1.689	3.445
45.000	0.434	0.532	1.306
0.000	0.239	0.049	0.582

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.492	0.101	0.956
135.000	0.561	0.627	1.073
90.000	0.058	1.607	4.536
45.000	0.587	0.617	2.688
0.000	0.532	0.155	2.620

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.206	0.101	0.437
135.000	0.378	0.627	1.165
90.000	0.058	1.607	3.381
45.000	0.434	0.617	1.042
0.000	0.239	0.155	0.964

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.445	0.436	3.483
135.000	0.371	0.644	3.342
90.000	0.143	3.487	4.951
45.000	0.355	0.450	2.029
0.000	0.380	0.267	1.858

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.143	0.169	1.132
135.000	0.255	0.523	1.630
90.000	0.066	1.571	3.445
45.000	0.249	0.584	1.306
0.000	0.136	0.123	0.582

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.445	0.446	0.956
135.000	0.371	0.656	1.073
90.000	0.143	3.416	4.536
45.000	0.355	0.825	2.688
0.000	0.380	0.318	2.620

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.143	0.151	0.437
135.000	0.255	0.607	1.165
90.000	0.066	1.493	3.381
45.000	0.249	0.567	1.042
0.000	0.136	0.212	0.964

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.379	0.000	3.497	3.505	0.774	0.000
135.000	0.564	0.000	3.635	3.673	0.675	0.000
90.000	0.097	0.000	4.995	4.995	3.522	0.000
45.000	0.605	0.000	1.574	1.569	0.485	0.000
0.000	0.442	0.000	1.935	1.938	0.542	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.379	0.000	1.104	1.143	0.774	0.000
135.000	0.564	0.000	1.157	1.237	0.675	0.000
90.000	0.097	0.000	4.300	4.299	3.522	0.000
45.000	0.605	0.000	2.873	2.929	0.485	0.000
0.000	0.442	0.000	2.714	2.745	0.542	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.437	0.535	3.497	3.518	0.774	0.000
135.000	0.461	0.467	3.635	3.652	0.675	0.000
90.000	0.181	2.433	4.995	4.720	3.522	0.000
45.000	0.432	0.335	1.574	1.549	0.485	0.000
0.000	0.358	0.375	1.935	1.971	0.542	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.437	0.535	1.104	1.177	0.774	0.000
135.000	0.461	0.466	1.157	1.195	0.675	0.000
90.000	0.181	2.433	4.300	3.959	3.522	0.000
45.000	0.432	0.335	2.873	2.912	0.485	0.000
0.000	0.358	0.375	2.714	2.717	0.542	0.000

WAMIT irregular seas response calcualtions: WAMIRGW
WAMIRGW input file: TACWAT1L.IRG
WAMIT .4 file: TACWATY1.4L
number of headings =   5 number of periods = 32   1
Reference length = 581.8300 Units conv   1.0000
Output file: TACWAT1L.OVT
Input units: FEET   Output units: FEET   Angles in degrees (deg/sec or deg/sec^2)

Sea State 3

Bretshnieder seas> H1/3=   4.100 To =   8.0000

Cut   1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	0.00000E+00	2.15781E+05	0.00000E+00	0.00000E+00	0.00000E+00	2.88954E+07
135.000	0.00000E+00	1.39413E+05	0.00000E+00	0.00000E+00	0.00000E+00	3.76352E+07
90.000	0.00000E+00	5.39836E+05	0.00000E+00	0.00000E+00	0.00000E+00	3.01830E+07
45.000	0.00000E+00	9.92710E+04	0.00000E+00	0.00000E+00	0.00000E+00	2.93119E+07
0.000	0.00000E+00	7.67382E+04	0.00000E+00	0.00000E+00	0.00000E+00	3.27638E+07

Sea State 4

Bretshnieder seas> H1/3=   8.200 To =   9.0000

Cut   1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	0.00000E+00	3.82327E+05	0.00000E+00	0.00000E+00	0.00000E+00	5.51372E+07
135.000	0.00000E+00	2.68096E+05	0.00000E+00	0.00000E+00	0.00000E+00	8.01946E+07
90.000	0.00000E+00	1.11605E+06	0.00000E+00	0.00000E+00	0.00000E+00	5.66888E+07
45.000	0.00000E+00	2.15574E+05	0.00000E+00	0.00000E+00	0.00000E+00	6.58141E+07
0.000	0.00000E+00	1.58192E+05	0.00000E+00	0.00000E+00	0.00000E+00	6.76044E+07

Sea State 5

Bretshnieder seas> H1/3=   13.120 To = 10.0000

Cut   1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	0.00000E+00	5.36542E+05	0.00000E+00	0.00000E+00	0.00000E+00	8.23715E+07
135.000	0.00000E+00	4.02843E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.29890E+08
90.000	0.00000E+00	1.74296E+06	0.00000E+00	0.00000E+00	0.00000E+00	8.17442E+07
45.000	0.00000E+00	3.46955E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.09710E+08
0.000	0.00000E+00	2.52130E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.05472E+08

APPENDIX D – T-ACS 5 AND LMSR USNS WATSON SURGE, SWAY, AND YAW RESTRAINT

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WAMIT irregular seas response calculations: WAMIRGW
 WAMIRGW input file:  TACWAT5.IRG
 WAMIT .4 file:  TACWATY5.4T
 number of headings =   5 number of periods =  32   2
 Reference length =  581.8300 Units conv    1.0000
 Output file:  TACWATY5.OVT
 Input units: FEET   Output units: FEET  Angles in degrees (deg/sec or
deg/sec^2)
```

SEA STATE 3

Bretshnieder seas> H1/3=   4.100 To =   8.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.048	0.074	0.322	0.213	0.146	0.003
135.000	0.059	0.075	0.303	0.177	0.129	0.021
90.000	0.022	0.213	0.724	0.753	0.043	0.009
45.000	0.069	0.054	0.221	0.111	0.137	0.020
0.000	0.062	0.042	0.196	0.092	0.158	0.008

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.031	0.040	0.131	0.046	0.040	0.003
135.000	0.053	0.061	0.187	0.077	0.057	0.021
90.000	0.008	0.383	0.790	0.113	0.010	0.009
45.000	0.050	0.052	0.156	0.068	0.060	0.020
0.000	0.033	0.023	0.111	0.037	0.046	0.008

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.023	0.034	0.792
135.000	0.033	0.066	0.686
90.000	0.011	0.363	1.084
45.000	0.037	0.064	0.337
0.000	0.023	0.012	0.375

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.023	0.034	0.245
135.000	0.033	0.066	0.333
90.000	0.011	0.363	0.883
45.000	0.037	0.064	0.212
0.000	0.023	0.012	0.118

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.023	0.035	0.189
135.000	0.033	0.079	0.199
90.000	0.011	0.340	0.964
45.000	0.037	0.074	0.569
0.000	0.023	0.043	0.617

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.023	0.035	0.087
135.000	0.033	0.079	0.232
90.000	0.011	0.340	0.852
45.000	0.037	0.074	0.218
0.000	0.023	0.043	0.219

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.112	0.157	0.792
135.000	0.087	0.126	0.686
90.000	0.032	0.838	1.084
45.000	0.093	0.096	0.337
0.000	0.110	0.063	0.375

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.026	0.043	0.245
135.000	0.033	0.074	0.333
90.000	0.015	0.302	0.883
45.000	0.029	0.085	0.212
0.000	0.025	0.030	0.118

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.112	0.162	0.189
135.000	0.087	0.152	0.199
90.000	0.032	0.816	0.964
45.000	0.093	0.107	0.569
0.000	0.110	0.074	0.617

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.026	0.041	0.087
135.000	0.033	0.073	0.232
90.000	0.015	0.282	0.852
45.000	0.029	0.061	0.218
0.000	0.025	0.056	0.219

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	0.892	0.892	0.256	0.000
135.000	0.000	0.000	0.901	0.901	0.175	0.000
90.000	0.000	0.000	1.337	1.337	0.860	0.000
45.000	0.000	0.000	0.400	0.400	0.096	0.000
0.000	0.000	0.000	0.462	0.462	0.125	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	0.259	0.259	0.256	0.000
135.000	0.000	0.000	0.344	0.344	0.175	0.000
90.000	0.000	0.000	1.094	1.094	0.860	0.000
45.000	0.000	0.000	0.724	0.724	0.096	0.000
0.000	0.000	0.000	0.720	0.720	0.125	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.114	0.177	0.892	0.906	0.256	0.000
135.000	0.102	0.121	0.901	0.907	0.175	0.000
90.000	0.034	0.594	1.337	1.280	0.860	0.000
45.000	0.114	0.066	0.400	0.410	0.096	0.000
0.000	0.127	0.086	0.462	0.480	0.125	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.114	0.177	0.259	0.302	0.256	0.000
135.000	0.102	0.121	0.344	0.349	0.175	0.000
90.000	0.034	0.594	1.094	1.014	0.860	0.000
45.000	0.114	0.066	0.724	0.734	0.096	0.000
0.000	0.127	0.086	0.720	0.730	0.125	0.000

SEA STATE 4

Bretshnieder seas> H1/3= 8.200 To = 9.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.166	0.156	0.733	0.409	0.367	0.007
135.000	0.194	0.229	0.749	0.408	0.333	0.066
90.000	0.042	0.618	1.769	1.798	0.087	0.017
45.000	0.243	0.165	0.556	0.276	0.354	0.068
0.000	0.195	0.103	0.468	0.226	0.385	0.017

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.112	0.075	0.296	0.093	0.099	0.007
135.000	0.200	0.180	0.434	0.181	0.163	0.066
90.000	0.016	0.928	1.816	0.242	0.017	0.017
45.000	0.189	0.161	0.366	0.156	0.164	0.068
0.000	0.123	0.047	0.257	0.084	0.110	0.017

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.087	0.074	1.933
135.000	0.133	0.210	1.774
90.000	0.023	0.897	2.738
45.000	0.151	0.216	0.950
0.000	0.096	0.023	0.943

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.087	0.074	0.598
135.000	0.133	0.210	0.841
90.000	0.023	0.896	2.010
45.000	0.151	0.216	0.620
0.000	0.096	0.023	0.277

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.087	0.068	0.471
135.000	0.133	0.262	0.505
90.000	0.023	0.848	2.470
45.000	0.151	0.257	1.436
0.000	0.096	0.094	1.472

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.087	0.068	0.219
135.000	0.133	0.262	0.598
90.000	0.023	0.848	1.960
45.000	0.151	0.257	0.535
0.000	0.096	0.094	0.530

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.256	0.288	1.933
135.000	0.216	0.311	1.774
90.000	0.067	1.977	2.738
45.000	0.216	0.234	0.950
0.000	0.247	0.151	0.943

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.066	0.101	0.598
135.000	0.081	0.229	0.841
90.000	0.029	0.793	2.010
45.000	0.090	0.260	0.620
0.000	0.072	0.069	0.277

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.256	0.301	0.471
135.000	0.216	0.358	0.505
90.000	0.067	1.933	2.470
45.000	0.216	0.347	1.436
0.000	0.247	0.173	1.472

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.066	0.090	0.219
135.000	0.081	0.242	0.598
90.000	0.029	0.748	1.960
45.000	0.090	0.220	0.535
0.000	0.072	0.128	0.530

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	2.076	2.076	0.496	0.000
135.000	0.000	0.000	2.133	2.133	0.391	0.000
90.000	0.000	0.000	3.021	3.021	2.028	0.000
45.000	0.000	0.000	0.903	0.903	0.243	0.000
0.000	0.000	0.000	1.087	1.087	0.300	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	0.617	0.617	0.496	0.000
135.000	0.000	0.000	0.732	0.732	0.391	0.000
90.000	0.000	0.000	2.538	2.538	2.028	0.000
45.000	0.000	0.000	1.680	1.680	0.243	0.000
0.000	0.000	0.000	1.630	1.630	0.300	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.272	0.343	2.076	2.099	0.496	0.000
135.000	0.241	0.270	2.133	2.144	0.391	0.000
90.000	0.067	1.401	3.021	2.867	2.028	0.000
45.000	0.262	0.168	0.903	0.926	0.243	0.000
0.000	0.290	0.207	1.087	1.125	0.300	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)
LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.272	0.342	0.617	0.681	0.496	0.000
135.000	0.241	0.270	0.732	0.739	0.391	0.000
90.000	0.067	1.401	2.538	2.337	2.028	0.000
45.000	0.262	0.168	1.680	1.704	0.243	0.000
0.000	0.290	0.207	1.630	1.648	0.300	0.000

SEA STATE 5

Bretshnieder seas> H1/3= 13.120 To = 10.0000

TACS 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.429	0.257	1.293	0.631	0.683	0.012
135.000	0.470	0.521	1.405	0.708	0.629	0.146
90.000	0.061	1.292	3.140	3.153	0.131	0.024
45.000	0.583	0.399	1.092	0.543	0.665	0.151
0.000	0.472	0.187	0.880	0.419	0.704	0.026

LMSR 6DOF LONGCRESTED SEAS RMS DISPLACEMENT:

WAMIT

Heading	SURGE	SWAY	HEAVE	ROLL	PITCH	YAW
180.000	0.328	0.110	0.514	0.147	0.201	0.012
135.000	0.502	0.411	0.793	0.307	0.341	0.146
90.000	0.027	1.716	3.131	0.396	0.024	0.024
45.000	0.481	0.386	0.690	0.262	0.336	0.151
0.000	0.352	0.072	0.447	0.141	0.214	0.026

Absolute Point 1 TACS connect #1

TACS xyz 195.500 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.270	0.120	3.492
135.000	0.357	0.501	3.335
90.000	0.037	1.688	4.952
45.000	0.397	0.513	1.994
0.000	0.291	0.036	1.835

Absolute Point 2 LMSR connect #1

LMSR xyz 195.500 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.270	0.120	1.133
135.000	0.357	0.501	1.632
90.000	0.037	1.688	3.446
45.000	0.397	0.513	1.308
0.000	0.291	0.036	0.581

Absolute Point 3 TACS connect #2

TACS xyz -151.300 -39.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST
180.000	0.270	0.101	0.969		
135.000	0.357	0.611	1.063		
90.000	0.037	1.608	4.533		
45.000	0.397	0.609	2.678		
0.000	0.291	0.149	2.611		

Absolute Point 4 LMSR connect #2

LMSR xyz -151.300 53.000 20.420 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.270	0.101	0.438
135.000	0.357	0.611	1.168
90.000	0.037	1.608	3.381
45.000	0.397	0.609	1.044
0.000	0.291	0.149	0.967

Absolute Point 5 TACS collision #1

TACS xyz 195.500 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.438	0.426	3.492
135.000	0.413	0.624	3.335
90.000	0.104	3.486	4.952
45.000	0.390	0.440	1.994
0.000	0.415	0.277	1.835

Absolute Point 6 LMSR collision #1

LMSR xyz 195.500 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.181	0.173	1.133
135.000	0.198	0.509	1.632
90.000	0.043	1.570	3.446
45.000	0.233	0.565	1.308
0.000	0.198	0.115	0.581

Absolute Point 7 TACS collision #2

TACS xyz -151.300 -39.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.438	0.449	0.969
135.000	0.413	0.650	1.063
90.000	0.104	3.417	4.533
45.000	0.390	0.816	2.678
0.000	0.415	0.305	2.611

Absolute Point 8 LMSR collision #2

LMSR xyz -151.300 53.000 60.000 (ref GDF origin)

LONGCRESTED SEAS RMS ABS DISPLACEMENT

Heading	X	Y	Z
180.000	0.181	0.147	0.438
135.000	0.198	0.593	1.168
90.000	0.043	1.494	3.381
45.000	0.233	0.560	1.044
0.000	0.198	0.211	0.967

Relative Point 1 Connect #1

TACS xyz 195.500 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	3.517	3.517	0.770	0.000
135.000	0.000	0.000	3.645	3.645	0.675	0.000
90.000	0.000	0.000	5.002	5.002	3.522	0.000
45.000	0.000	0.000	1.557	1.557	0.488	0.000
0.000	0.000	0.000	1.909	1.909	0.541	0.000

Relative Point 2 Connect #2

TACS xyz -151.300 -39.000 20.420 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 20.420 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.000	0.000	1.120	1.120	0.770	0.000
135.000	0.000	0.000	1.174	1.174	0.675	0.000
90.000	0.000	0.000	4.290	4.290	3.522	0.000
45.000	0.000	0.000	2.873	2.873	0.488	0.000
0.000	0.000	0.000	2.713	2.713	0.541	0.000

Relative Point 3 Collision #1

TACS xyz 195.500 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz 195.500 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.470	0.532	3.517	3.545	0.770	0.000
135.000	0.419	0.467	3.645	3.660	0.675	0.000
90.000	0.101	2.434	5.002	4.729	3.522	0.000
45.000	0.448	0.337	1.557	1.587	0.488	0.000
0.000	0.489	0.374	1.909	1.971	0.541	0.000

Relative Point 4 Collision #2

TACS xyz -151.300 -39.000 60.000 1.000 (ref GDF origin)
LMSR xyz -151.300 53.000 60.000 1.000 (ref GDF origin)

LONGCRESTED SEAS RMS REL DISPLACEMENT

Heading	X	Y	Z	ELONGATION	TWIST	
180.000	0.470	0.532	1.120	1.200	0.770	0.000
135.000	0.419	0.466	1.174	1.193	0.675	0.000
90.000	0.101	2.433	4.290	3.946	3.522	0.000
45.000	0.448	0.337	2.873	2.919	0.488	0.000
0.000	0.489	0.374	2.713	2.738	0.541	0.000

WAMIT irregular seas response calculations: WAMIRGW
WAMIRGW input file: TACWAT5L.IRG
WAMIT .4 file: TACWATY5.4L
number of headings = 5 number of periods = 32 1
Reference length = 581.8300 Units conv 1.0000
Output file: TACWAT5L.OVT
Input units: FEET Output units: FEET Angles in degrees (deg/sec or deg/sec^2)

Sea State 3

Bretshnieder seas> H1/3= 4.100 To = 8.0000

Cut 1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	5.40646E+04	2.13429E+05	0.00000E+00	0.00000E+00	0.00000E+00	2.94582E+07
135.000	4.32458E+04	1.37737E+05	0.00000E+00	0.00000E+00	0.00000E+00	3.82864E+07
90.000	4.23266E+04	5.39770E+05	0.00000E+00	0.00000E+00	0.00000E+00	3.09860E+07
45.000	4.58341E+04	9.95163E+04	0.00000E+00	0.00000E+00	0.00000E+00	2.94248E+07
0.000	4.93218E+04	7.61573E+04	0.00000E+00	0.00000E+00	0.00000E+00	3.25878E+07

Sea Sate 4

Bretshnieder seas> H1/3= 8.200 To = 9.0000

Cut 1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	1.14332E+05	3.78347E+05	0.00000E+00	0.00000E+00	0.00000E+00	5.57073E+07
135.000	1.19056E+05	2.65190E+05	0.00000E+00	0.00000E+00	0.00000E+00	8.13049E+07
90.000	7.51758E+04	1.11507E+06	0.00000E+00	0.00000E+00	0.00000E+00	5.81698E+07
45.000	1.30425E+05	2.15758E+05	0.00000E+00	0.00000E+00	0.00000E+00	6.57046E+07
0.000	1.17515E+05	1.57230E+05	0.00000E+00	0.00000E+00	0.00000E+00	6.66645E+07

Sea State 5

Bretshnieder seas> H1/3= 13.120 To = 10.0000

Cut 1 6DOF LONGCRESTED SEAS RMS WAVE LOADS:
WAMIT

Heading	Fx	Fy	Fz	Mx	My	Mz
180.000	1.95944E+05	5.31218E+05	0.00000E+00	0.00000E+00	0.00000E+00	8.23721E+07
135.000	2.41874E+05	3.98737E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.31306E+08
90.000	1.04775E+05	1.74050E+06	0.00000E+00	0.00000E+00	0.00000E+00	8.38536E+07
45.000	2.60786E+05	3.47109E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.09130E+08
0.000	2.13217E+05	2.50916E+05	0.00000E+00	0.00000E+00	0.00000E+00	1.03268E+08

Appendix E – SAIC plots

Irregular Seas Cases with CG's Aligned

Although these cases included 3 sea states and 2 speeds, only Sea State 5 at 8 knots is plotted here. Figure 1 through Figure 4 show the relative motion between deck edge points.

Relative motion is the change in distance between two corresponding deck edge points on the two ships.

Figure 5 shows the roll response for the first ship, Figure 6 for the second or sheltered ship. Figure 11 through Figure 16 show the sway force, and Figure 17 through Figure 22 show the yaw moment.

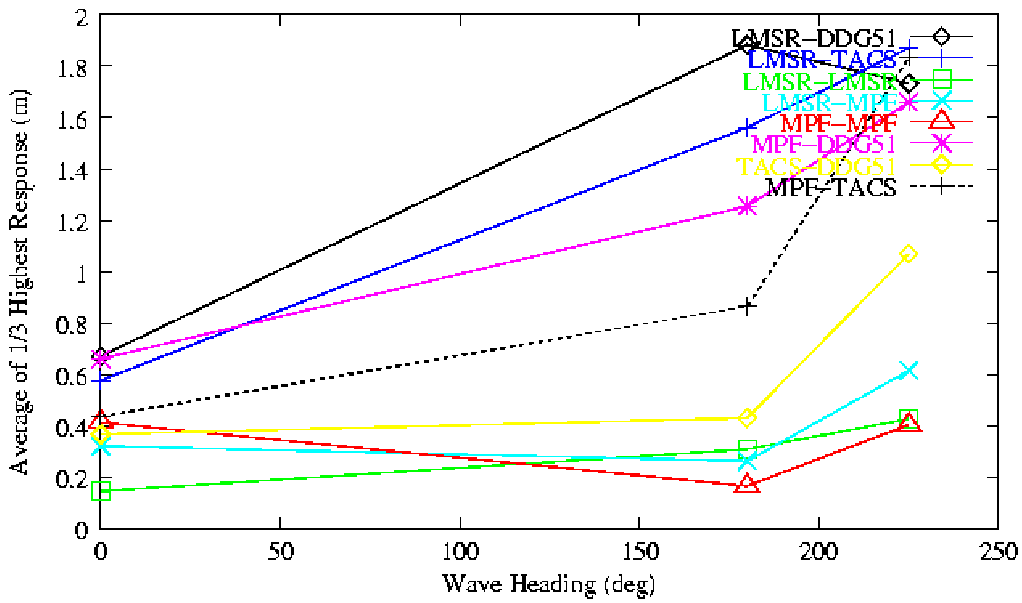

Figure 1 Relative Motion in Sea State 5 at 8 knots for Point 1

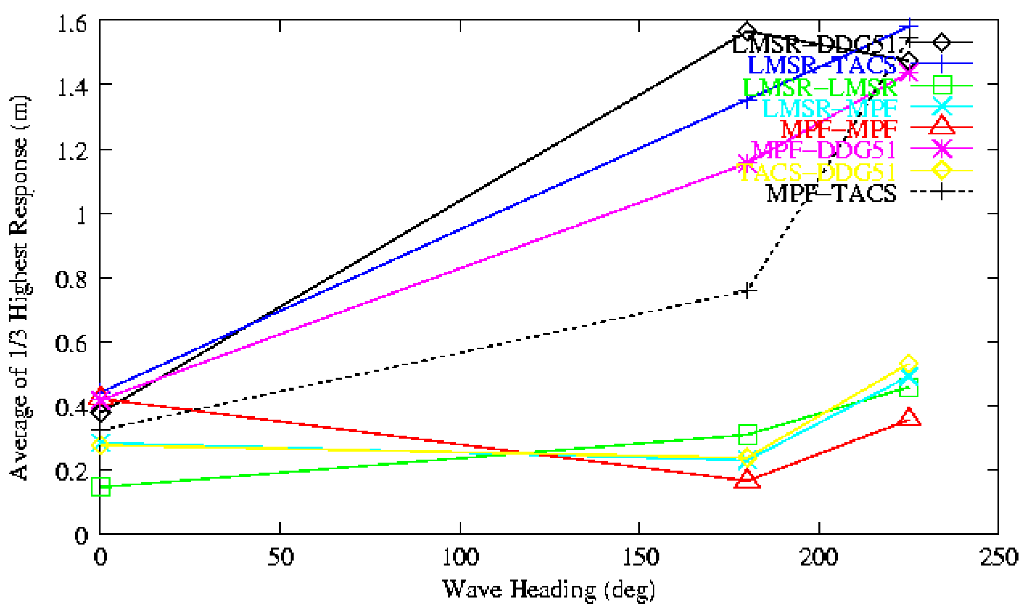

Figure 2 Relative Motion in Sea State 5 at 8 knots for Point 2

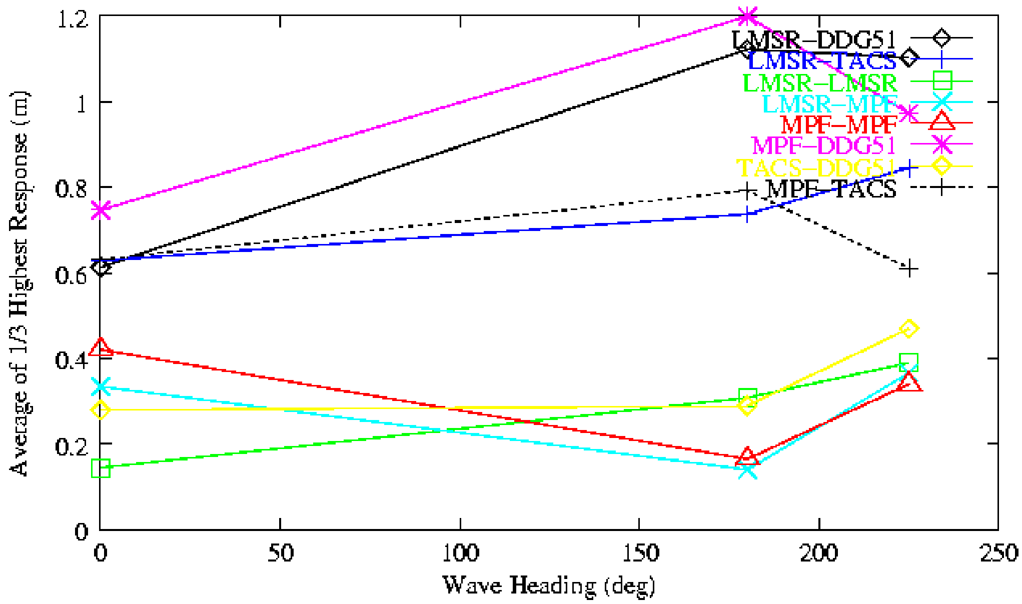

Figure 3 Relative Motion in Sea State 5 at 8 knots for Point 3

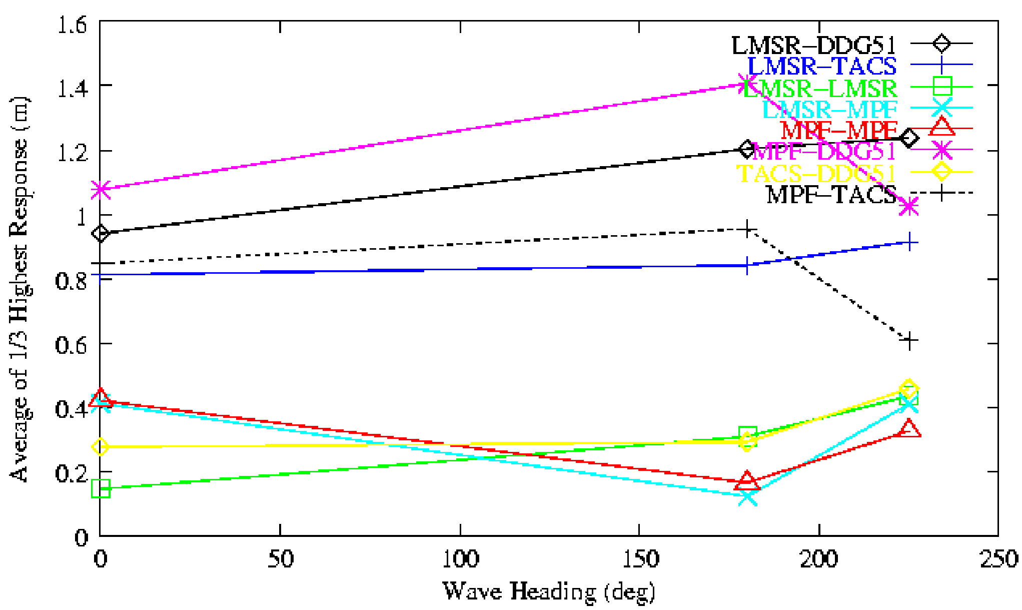

Figure 4 Relative Motion in Sea State 5 at 8 knots for Point 4

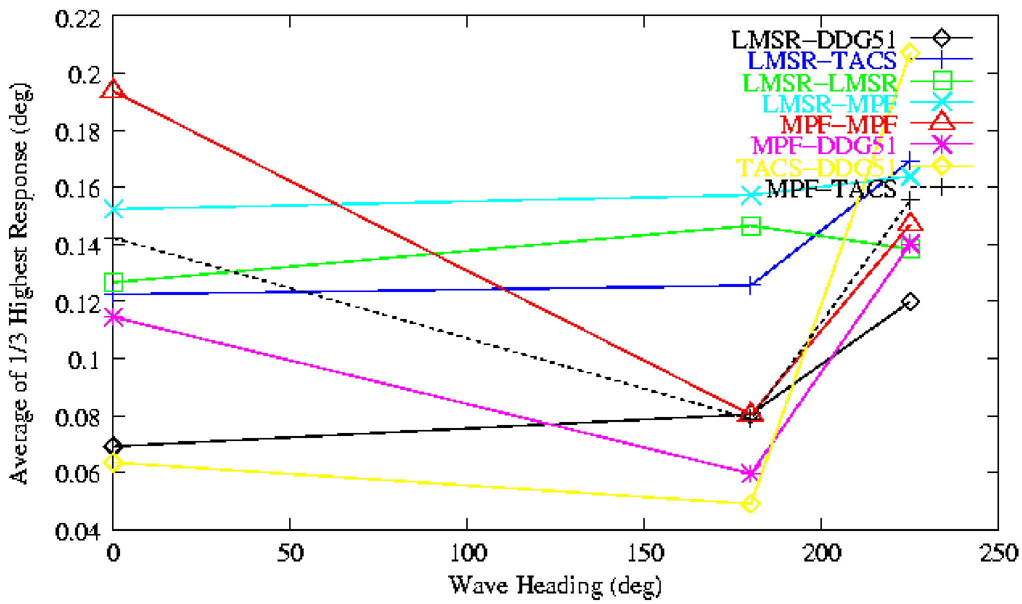

Figure 5 Roll Angle in Sea State 3 at 8 knots for Ship 1

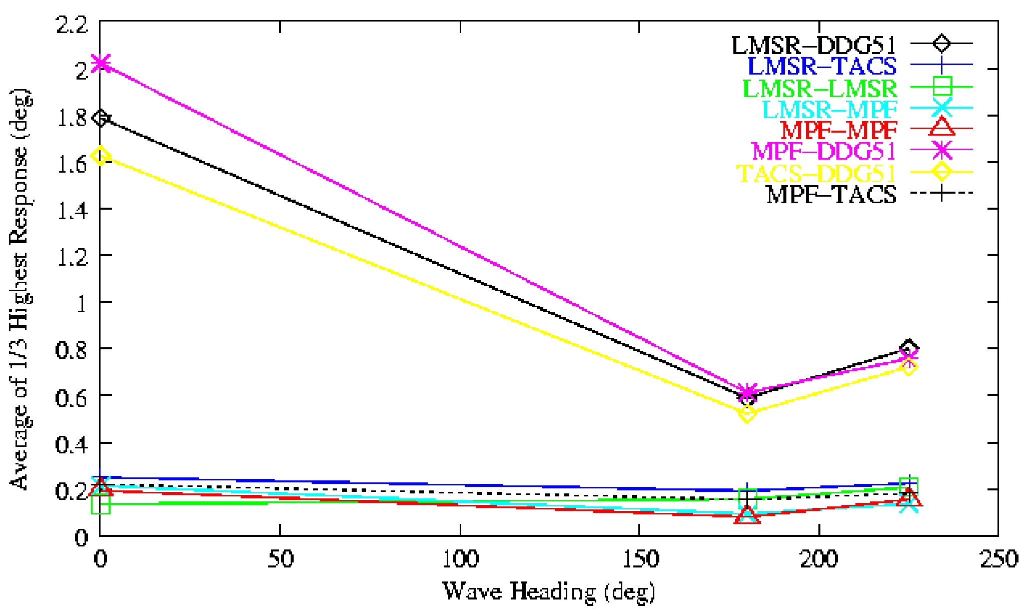

Figure 6 Roll Angle in Sea State 3 at 8 knots for Ship 2

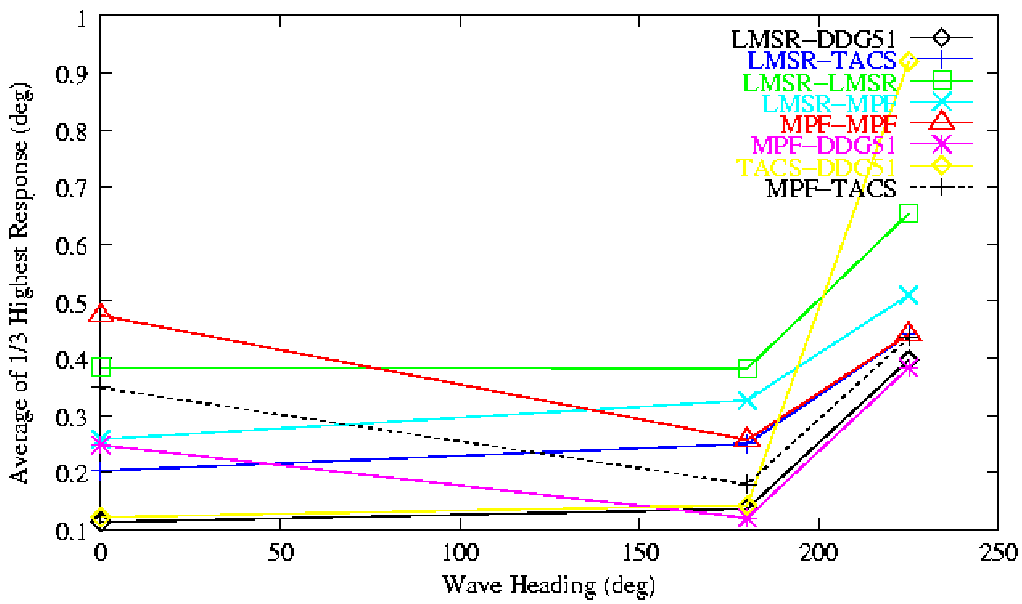

Figure 7 Roll Angle in Sea State 4 at 8 knots for Ship 1

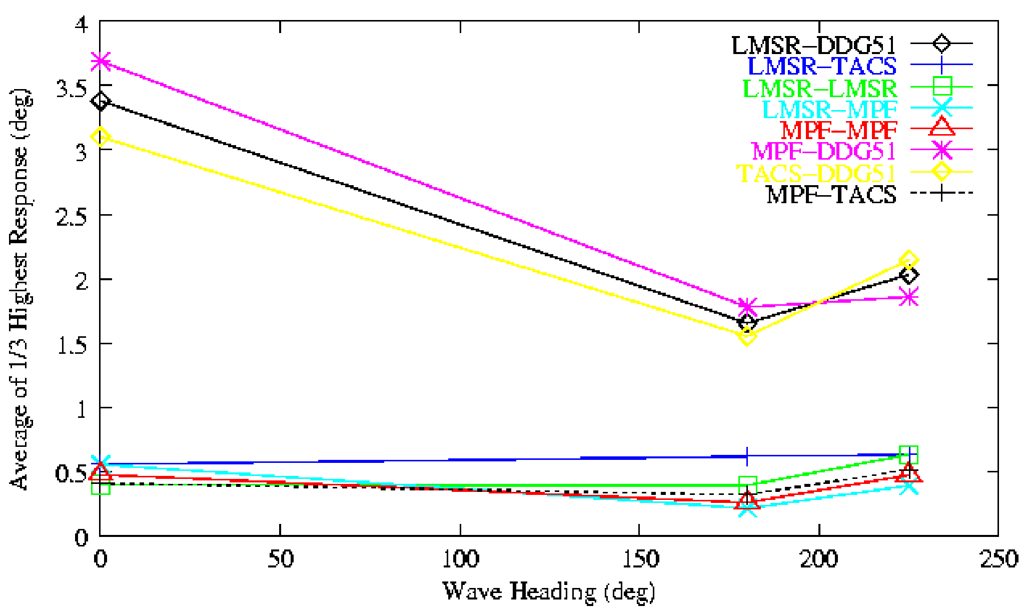

Figure 8 Roll Angle in Sea State 4 at 8 knots for Ship 2

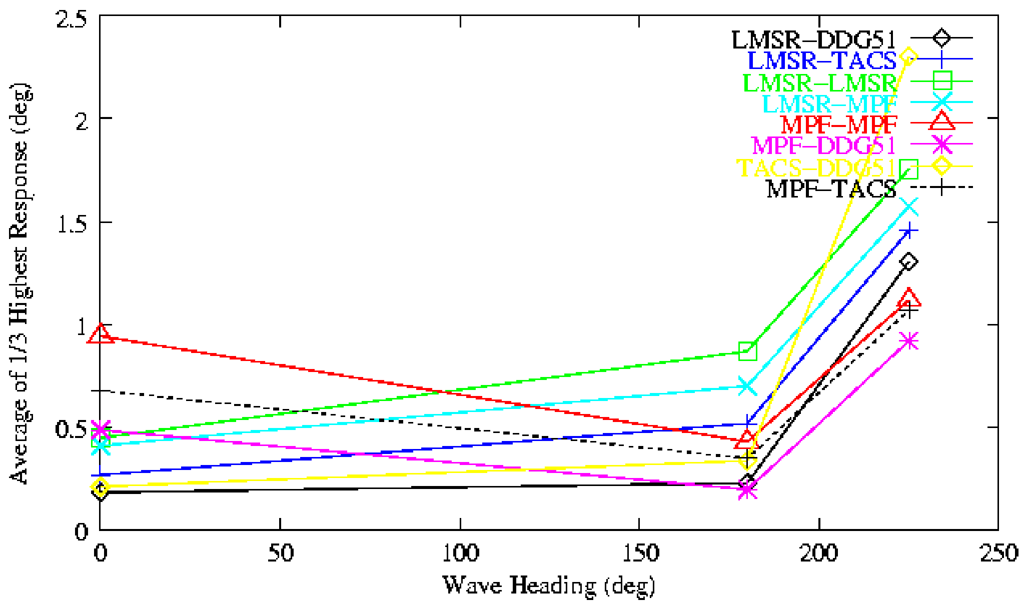

Figure 9 Roll Angle in Sea State 5 at 8 knots for Ship 1

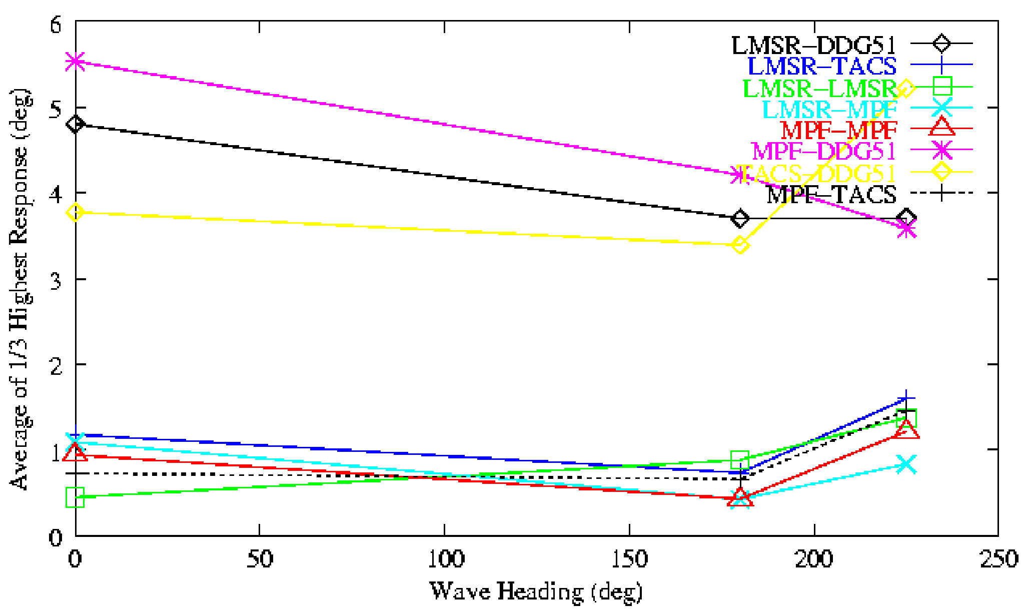

Figure 10 Roll Angle in Sea State 5 at 8 knots for Ship 2

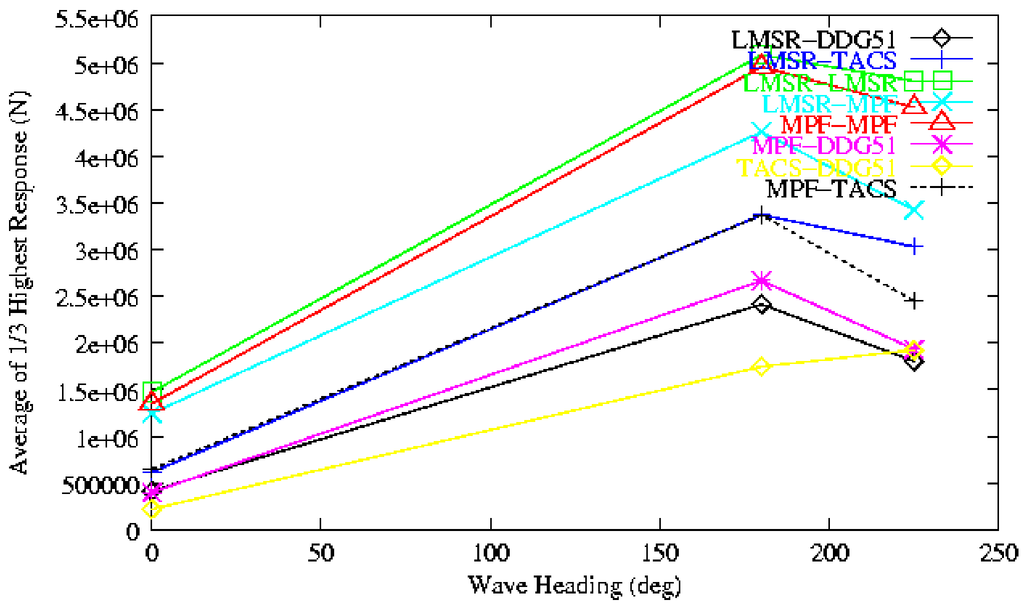

Figure 11 Sway Force in Sea State 3 at 8 knots for Ship 1

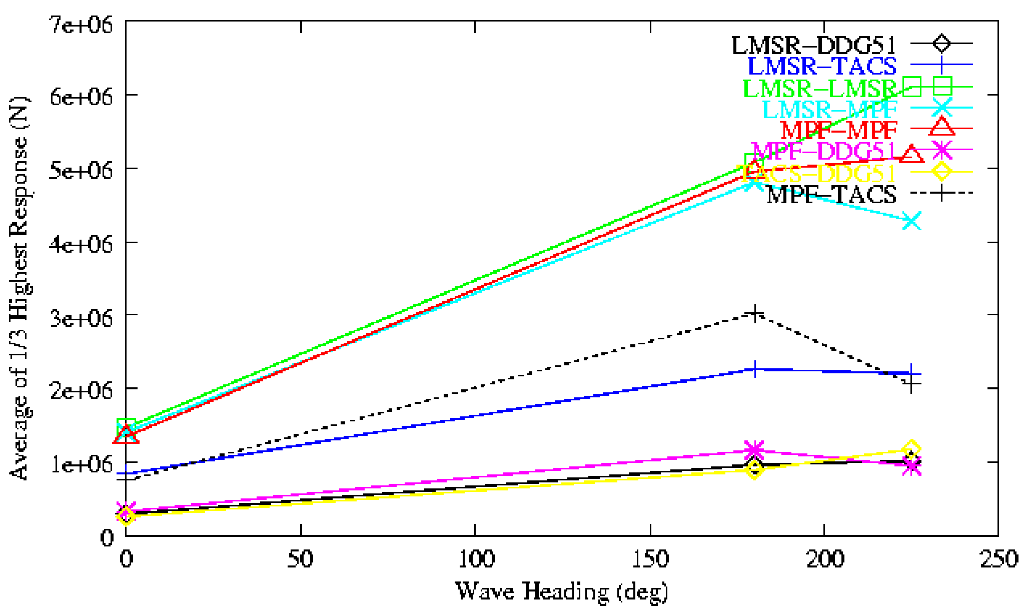

Figure 12 Sway Force in Sea State 3 at 8 knots for Ship 2

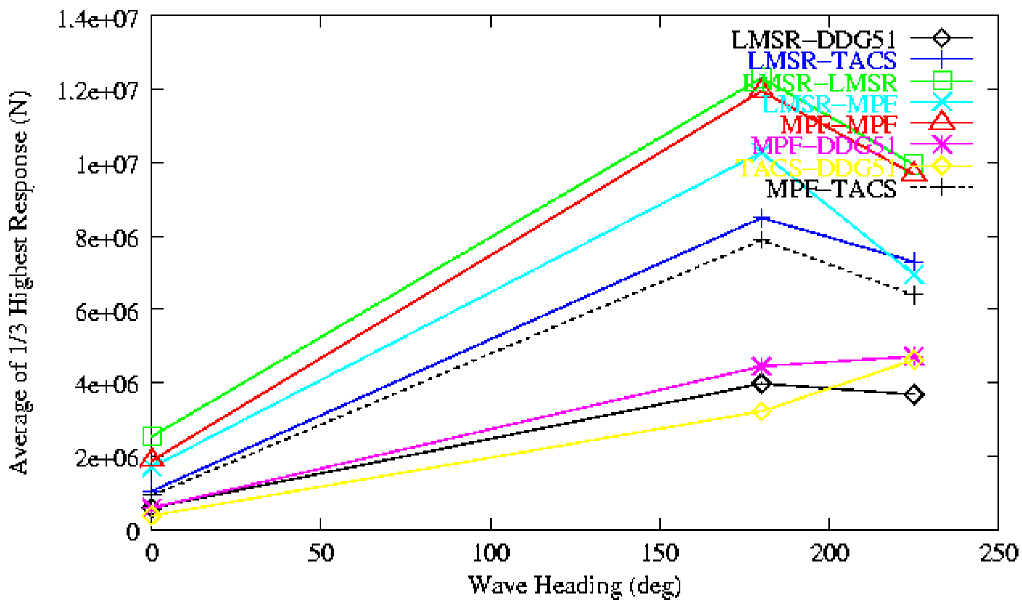

Figure 13 Sway Force in Sea State 4 at 8 knots for Ship 1

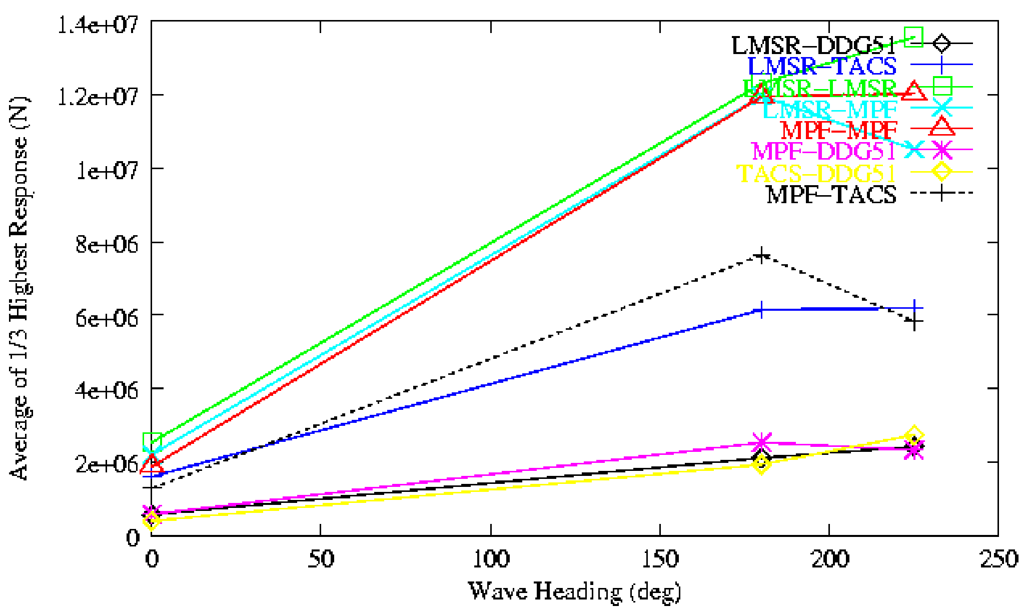

Figure 14 Sway Force in Sea State 4 at 8 knots for Ship 2

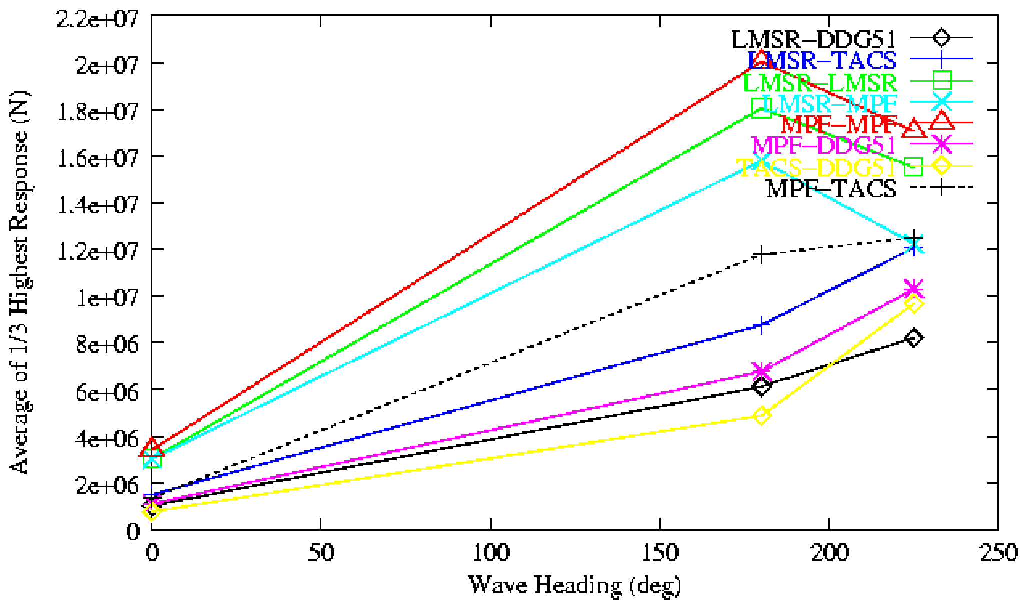

Figure 15 Sway Force in Sea State 5 at 8 knots for Ship 1

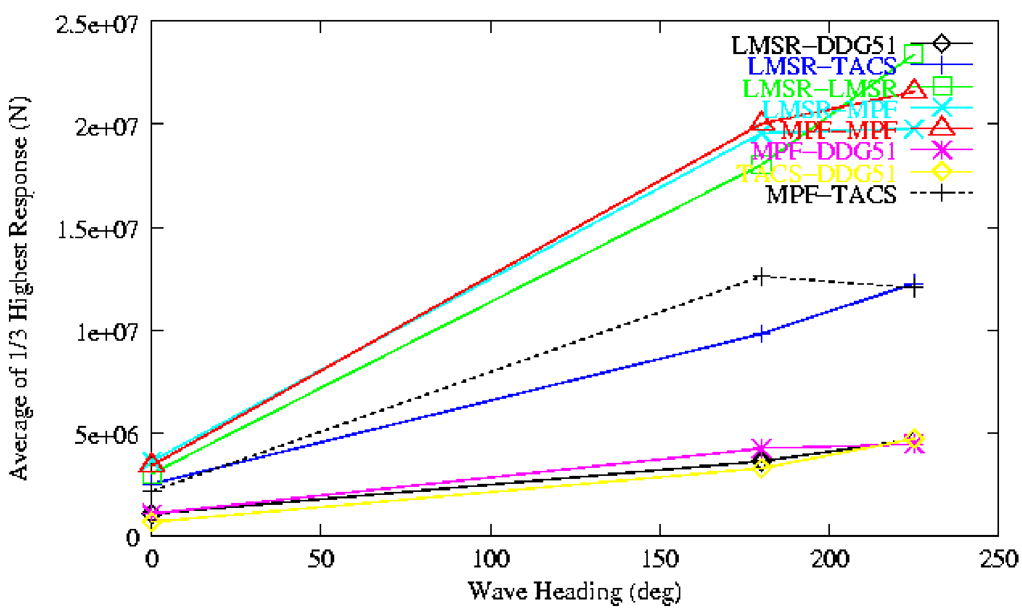

Figure 16 Sway Force in Sea State 5 at 8 knots for Ship 2

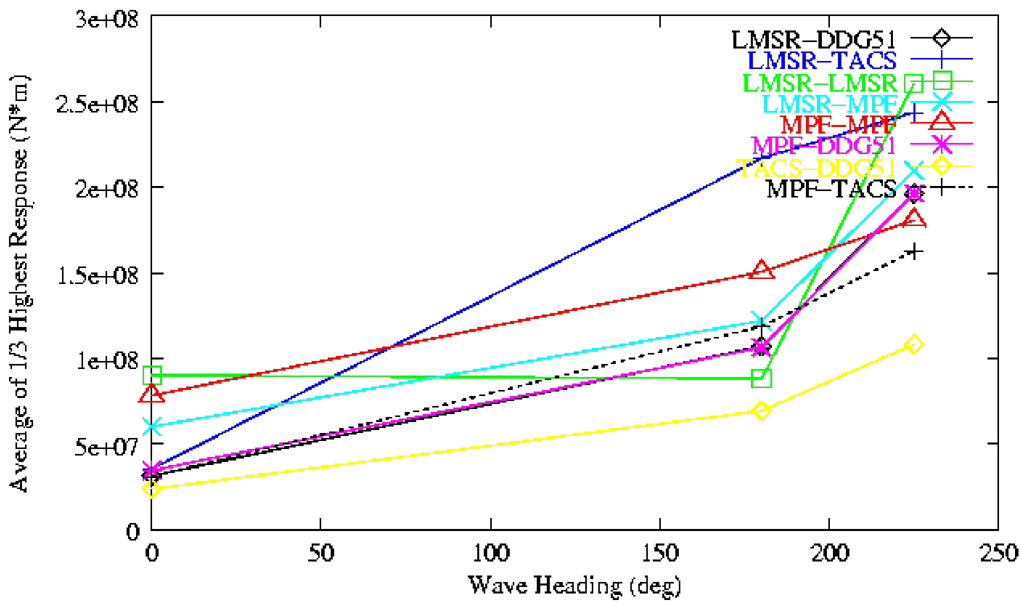

Figure 17 Yaw Moment in Sea State 3 at 8 knots for Ship 1

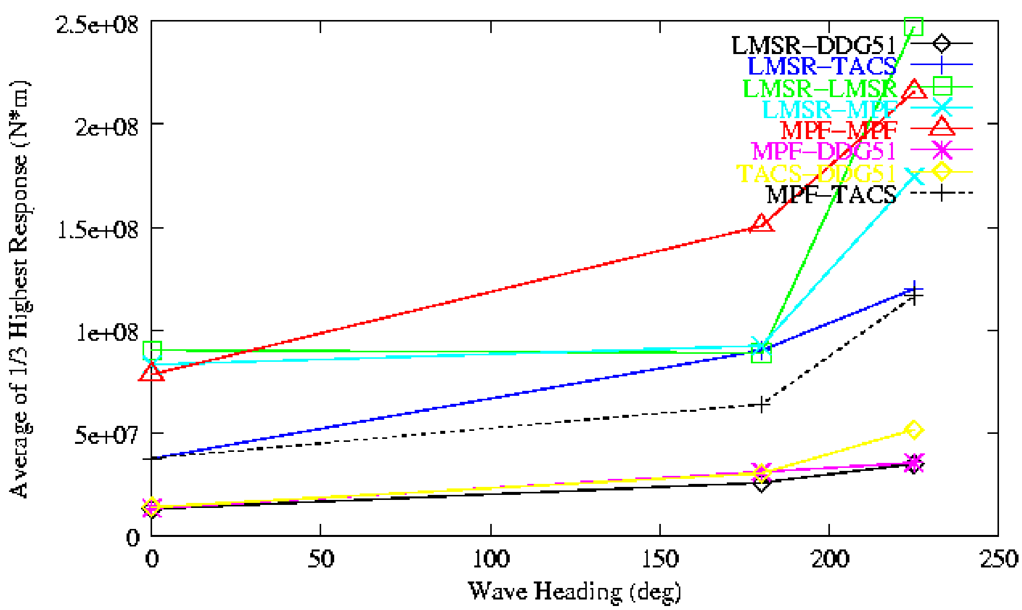

Figure 18 Yaw Moment in Sea State 3 at 8 knots for Ship 2

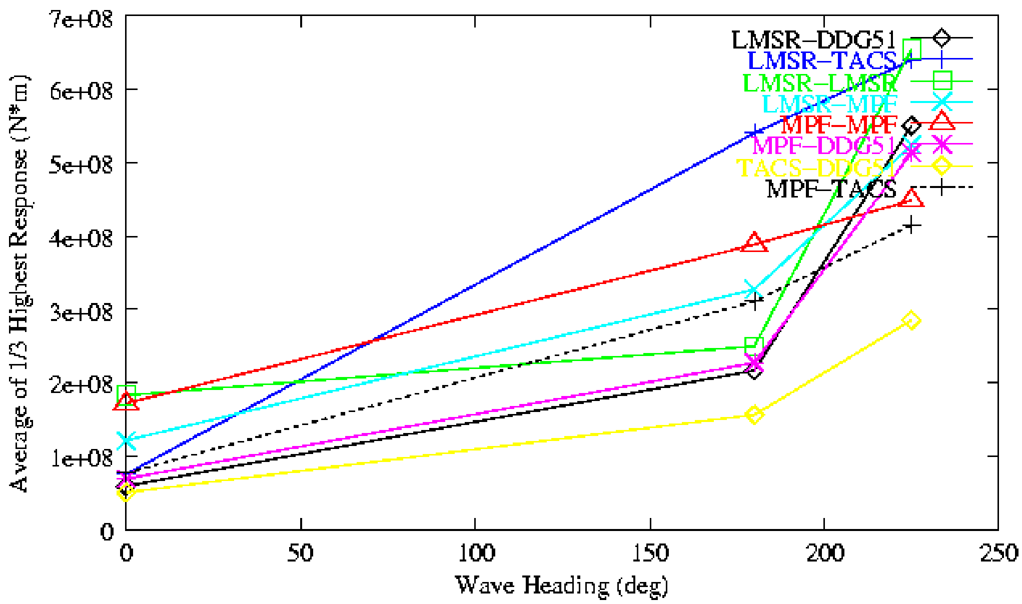

Figure 19 Yaw Moment in Sea State 4 at 8 knots for Ship 1

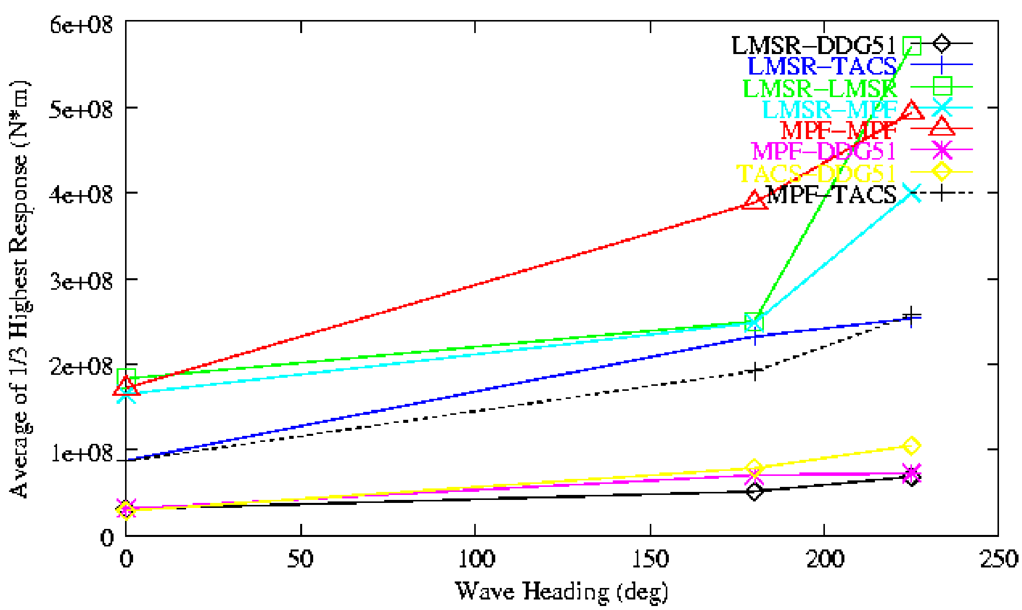

Figure 20 Yaw Moment in Sea State 4 at 8 knots for Ship 2

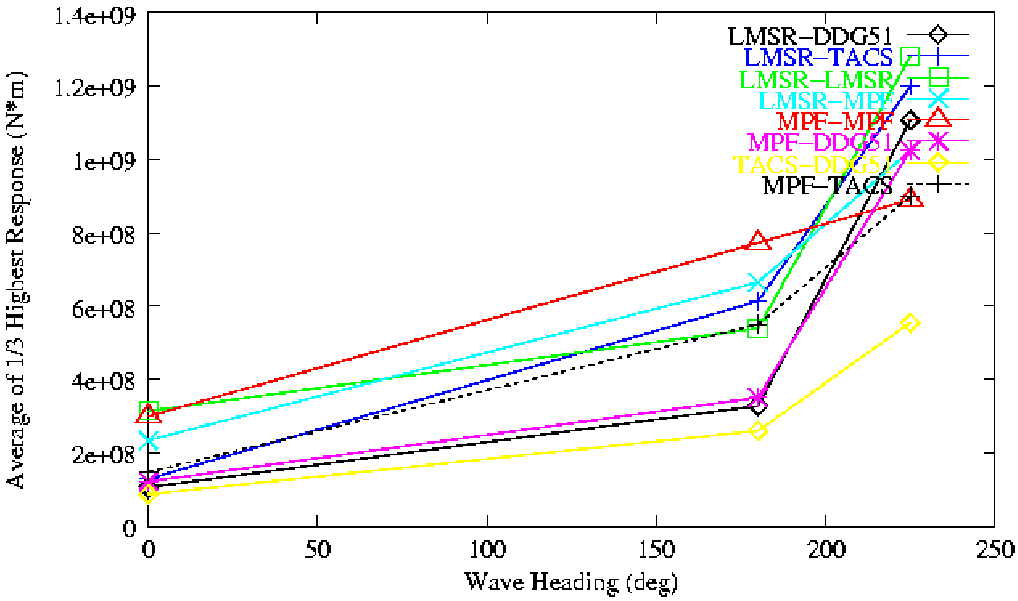

Figure 21 Yaw Moment in Sea State 5 at 8 knots for Ship 1

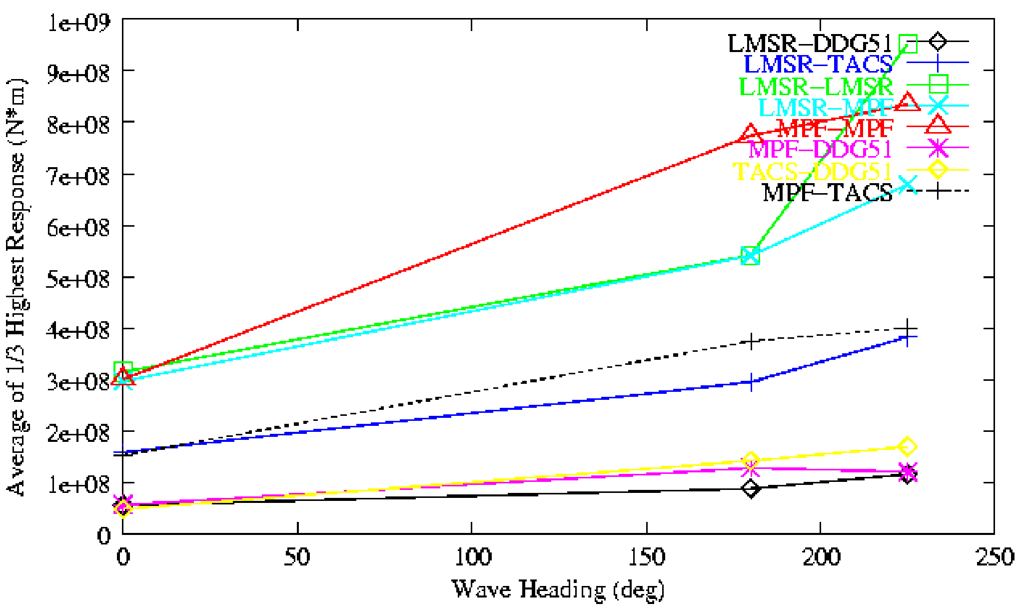

Figure 22 Yaw Moment in Sea State 5 at 8 knots for Ship 2

Irregular Seas Cases with Station 15 Aligned

These cases included 2 sea states and 2 speeds, but only Sea State 5 at 8 knots are plotted here.

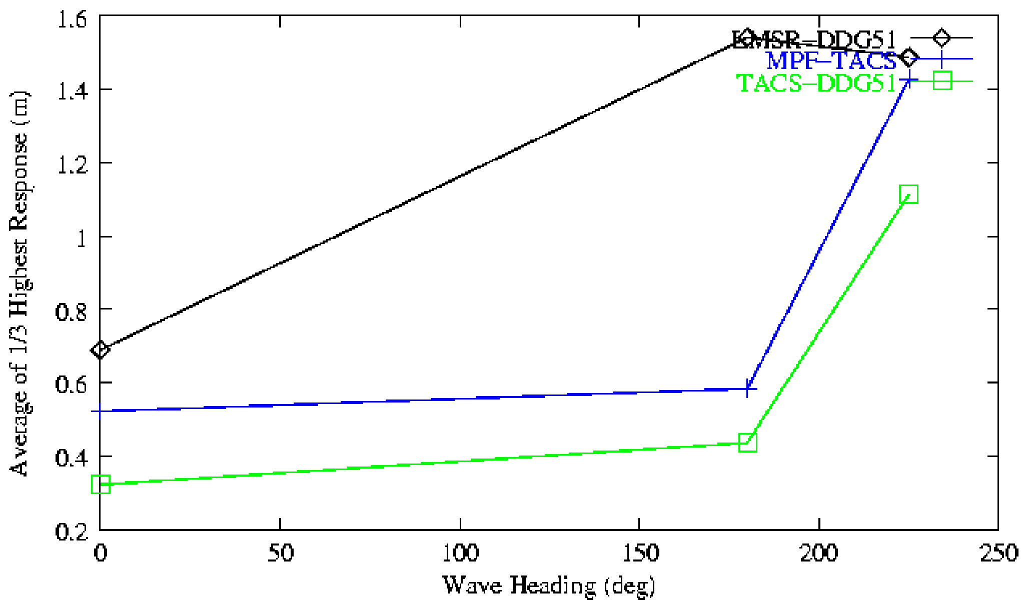

Figure 23 Relative Motion with Station 15 Aligned in Sea State 5 at 8 knots for Point 1

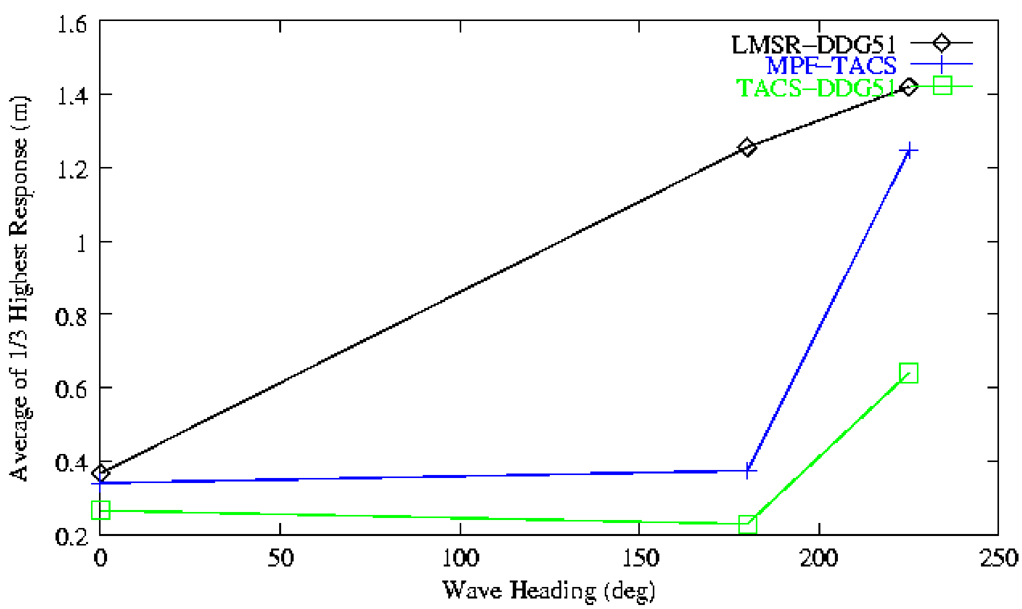

Figure 24 Relative Motion with Station 15 Aligned in Sea State 5 at 8 knots for Point 2

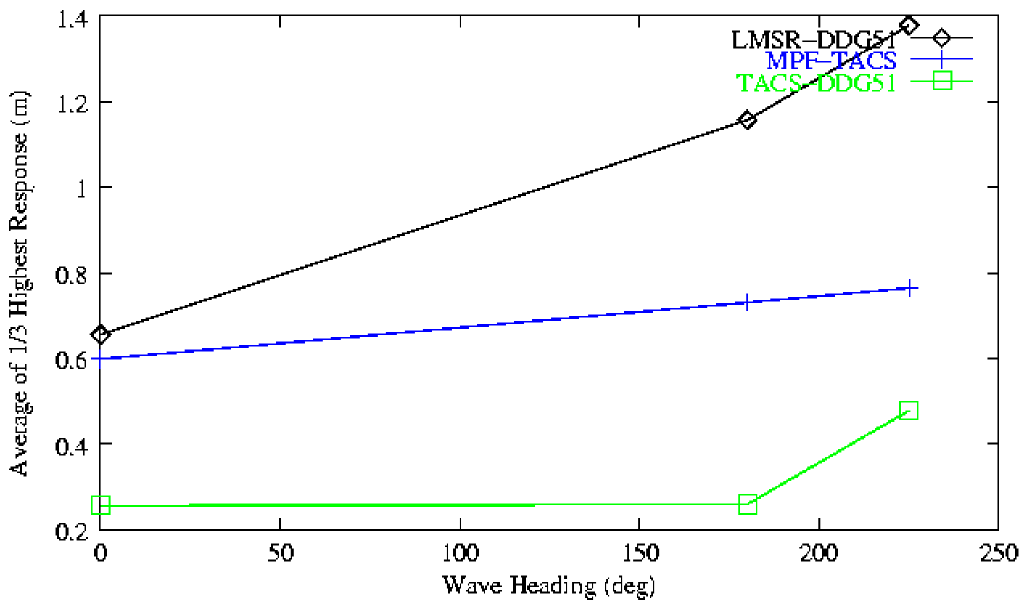

Figure 25 Relative Motion with Station 15 Aligned in Sea State 5 at 8 knots for Point 3

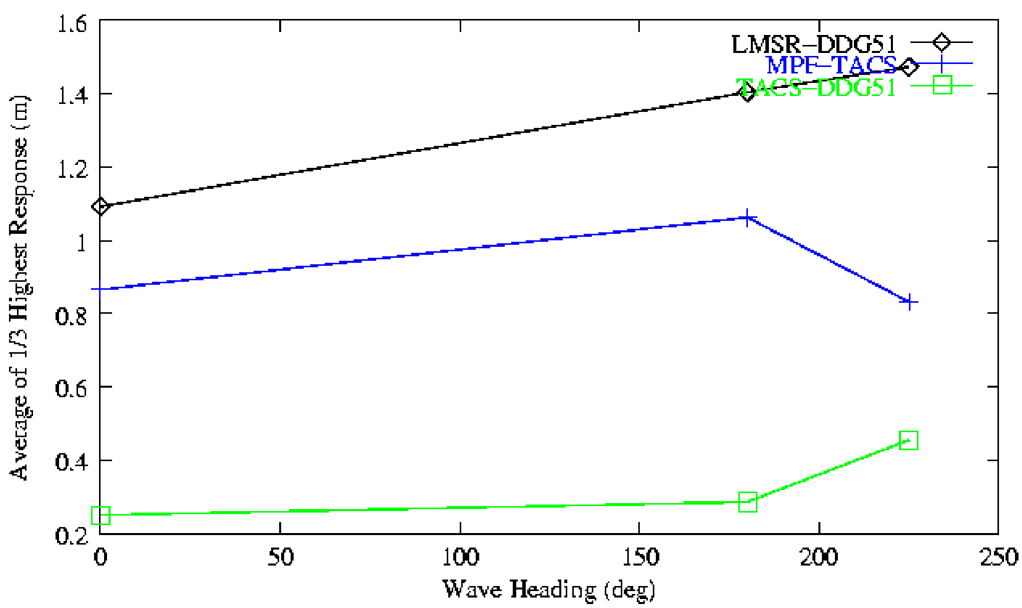

Figure 26 Relative Motion with Station 15 Aligned in Sea State 5 at 8 knots for Point 4

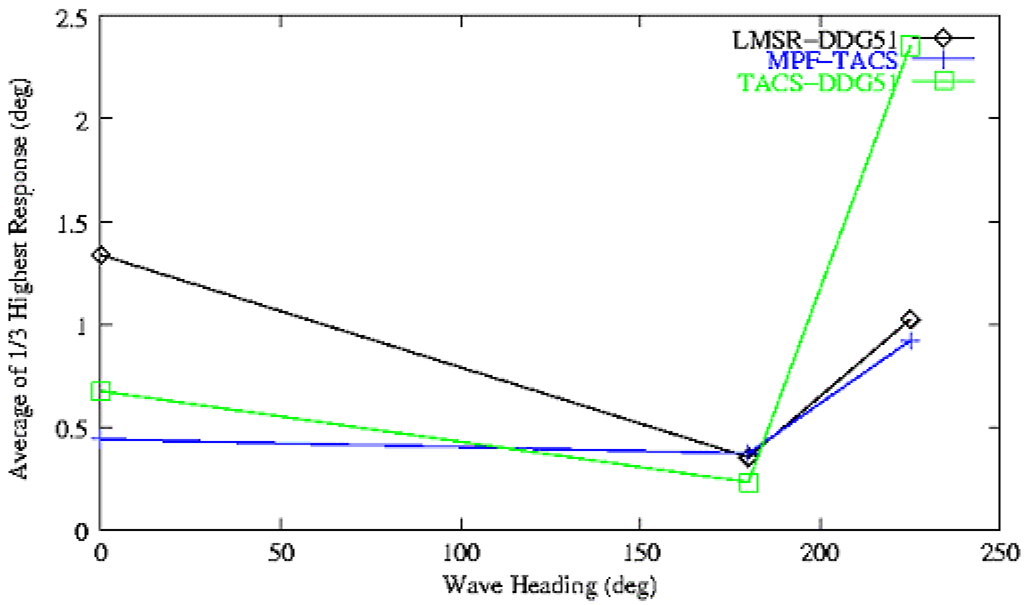

Figure 27 Roll Angle with Station 15 Aligned in Sea State 5 at 16 knots for Ship 1

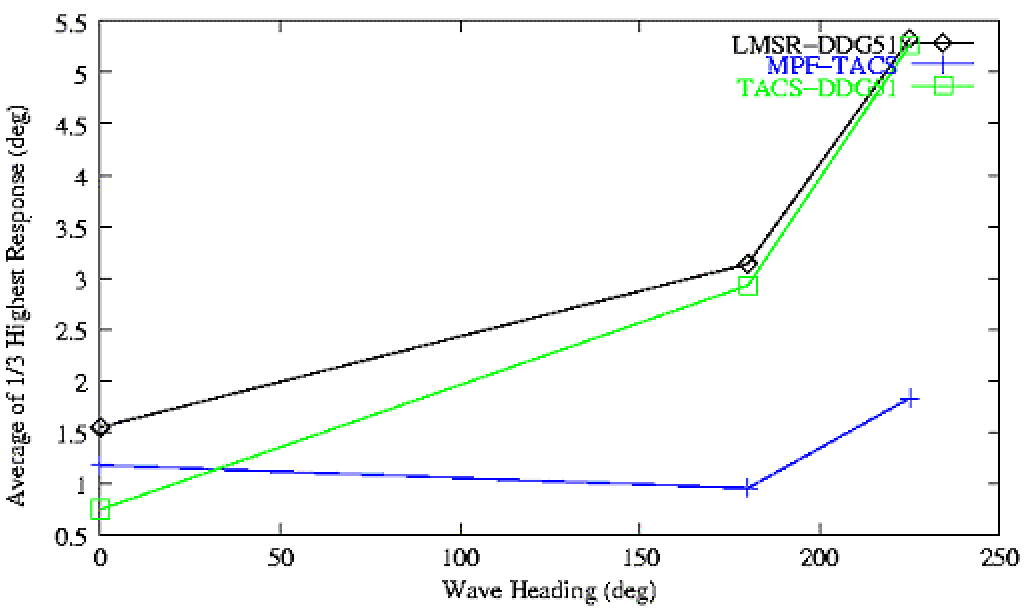

Figure 28 Roll Angle with Station 15 Aligned in Sea State 5 at 16 knots for Ship 2

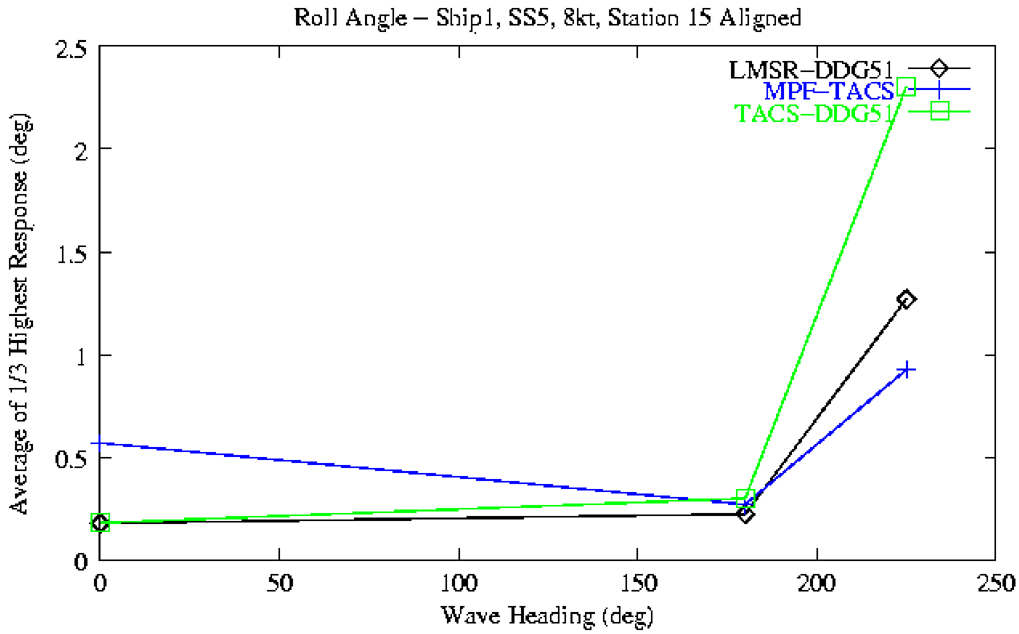

Figure 29 Roll Angle with Station 15 Aligned in Sea State 5 at 8 knots for Ship 1

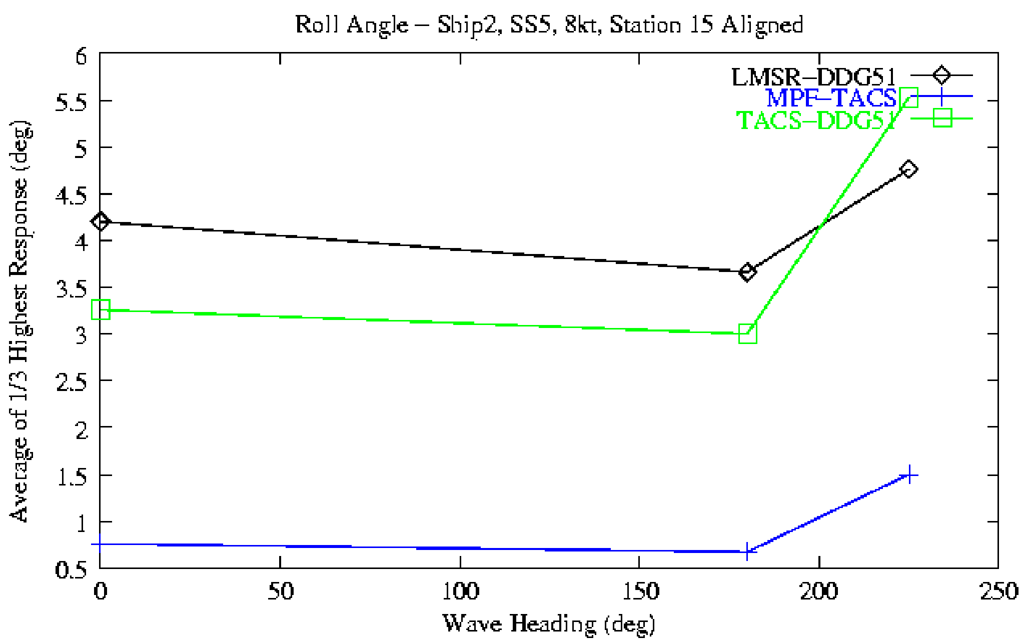

Figure 30 Roll Angle with Station 15 Aligned in Sea State 5 at 8 knots for Ship 2

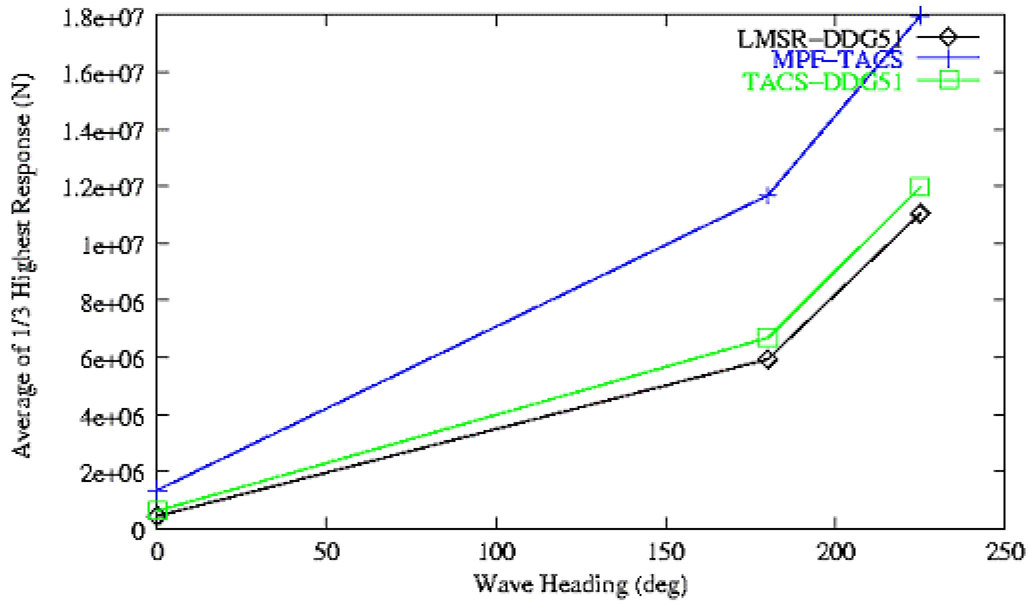

Figure 31 Sway Force with Station 15 Aligned in Sea State 5 at 16 knots for Ship 1

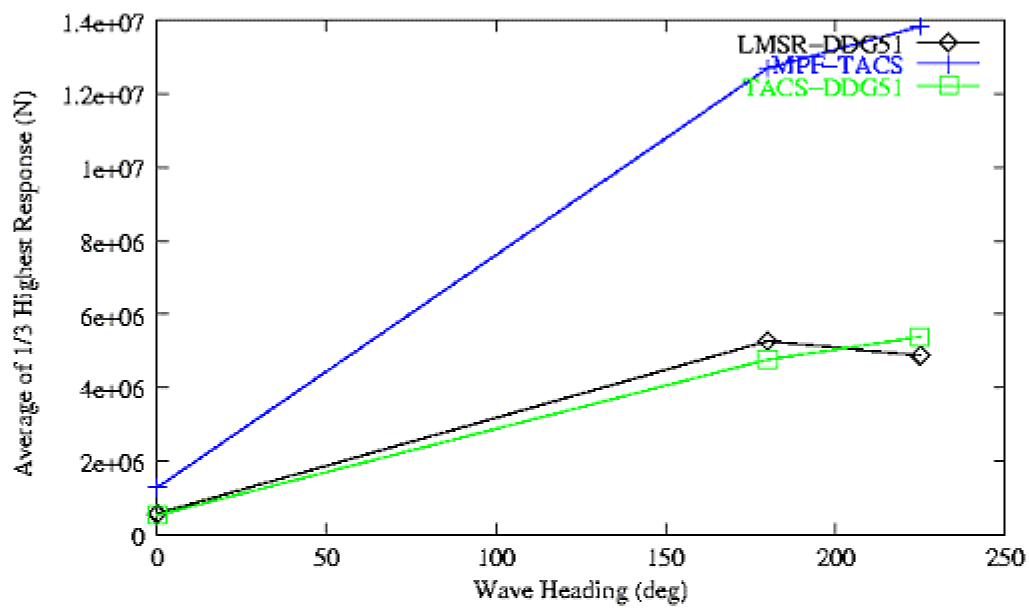

Figure 32 Sway Force with Station 15 Aligned in Sea State 5 at 16 knots for Ship 2

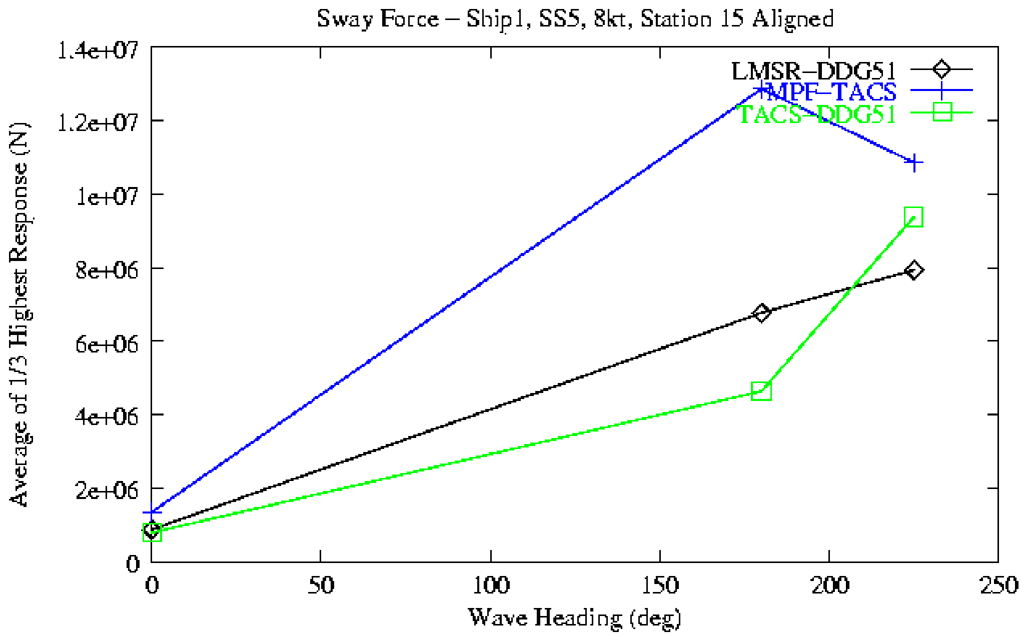

Figure 33 Sway Force with Station 15 Aligned in Sea State 5 at 8 knots for Ship 1

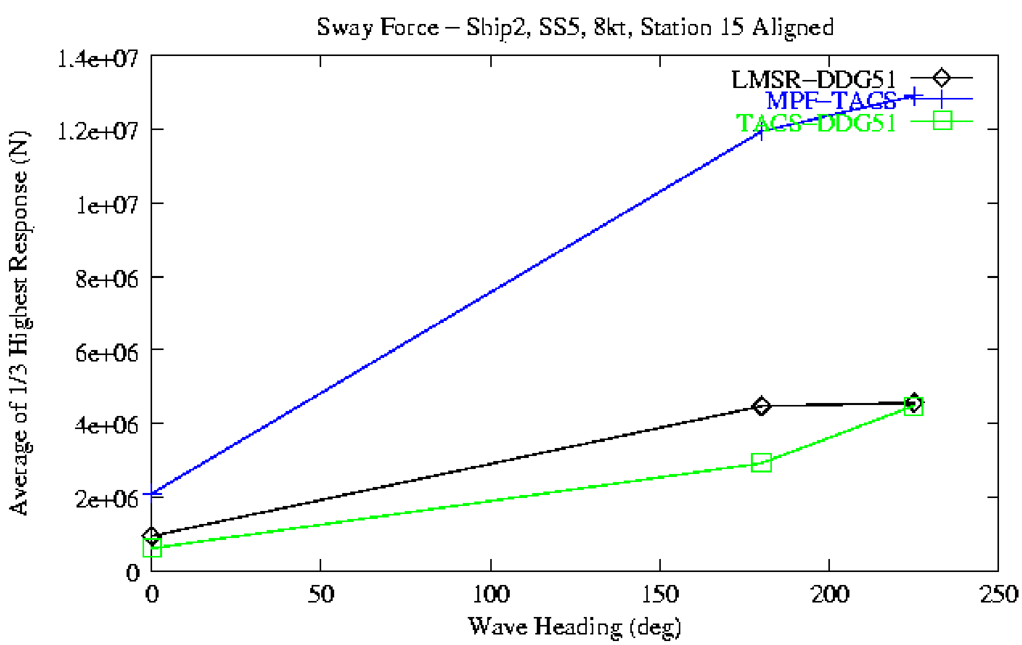

Figure 34 Sway Force with Station 15 Aligned in Sea State 5 at 8 knots for Ship 2

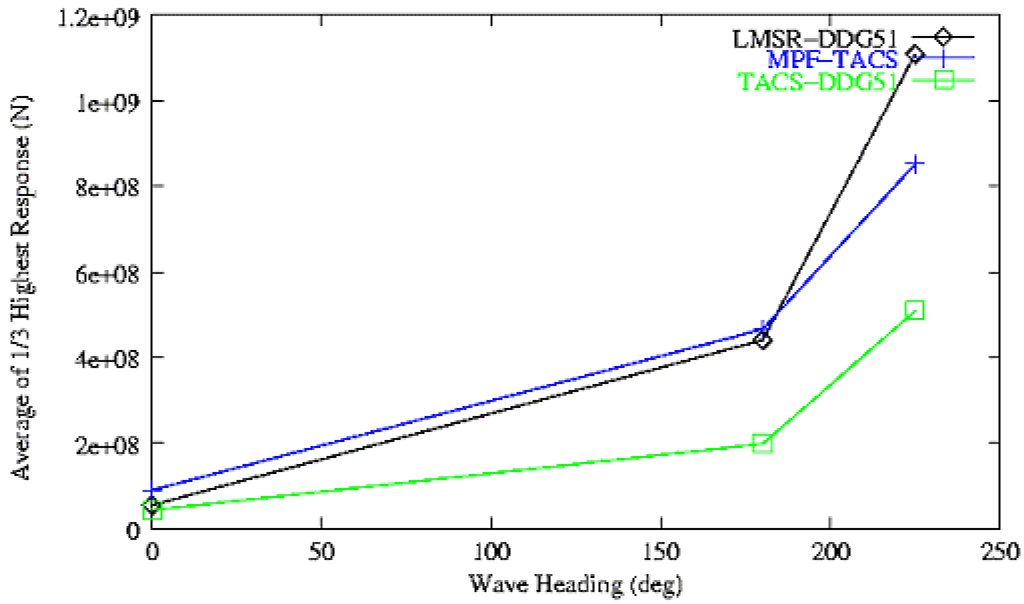

Figure 35 Yaw Moment with Station 15 Aligned in Sea State 5 at 16 knots for Ship 1

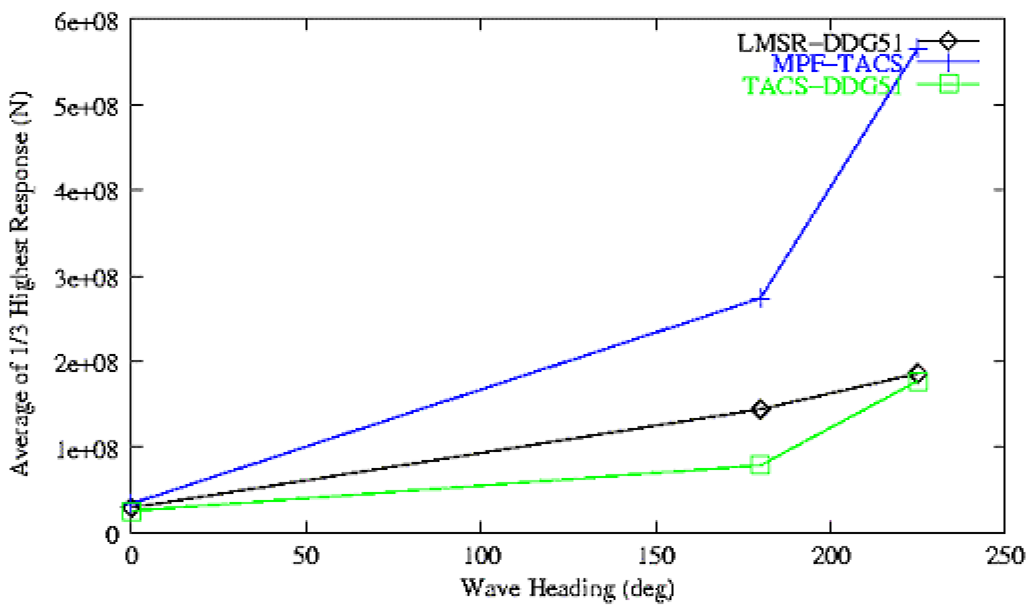

Figure 36 Yaw Moment with Station 15 Aligned in Sea State 5 at 16 knots for Ship 2

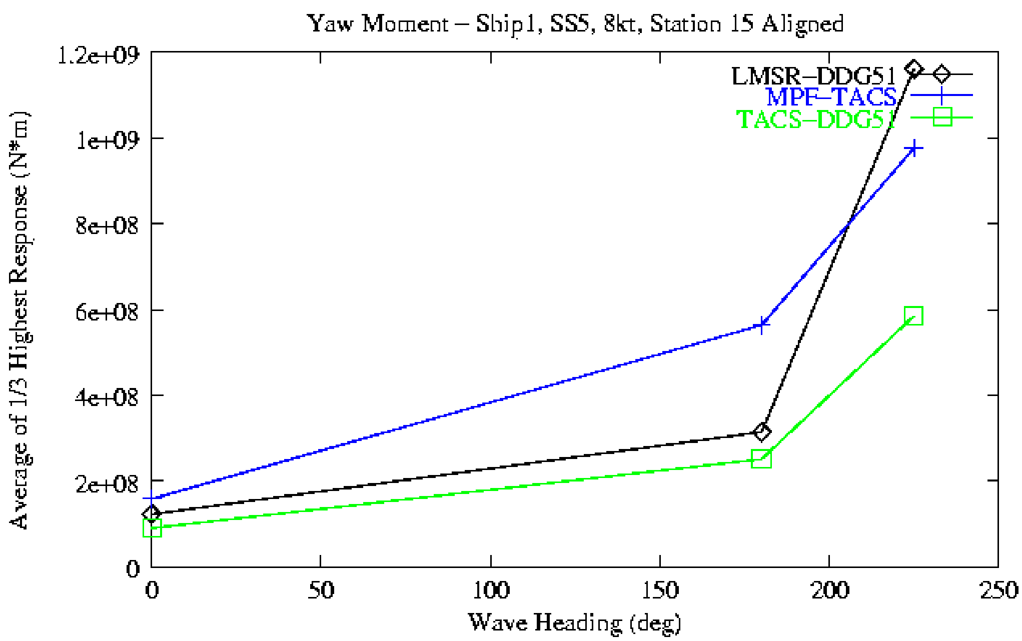

Figure 37 Yaw Moment with Station 15 Aligned in Sea State 5 at 8 knots for Ship 1

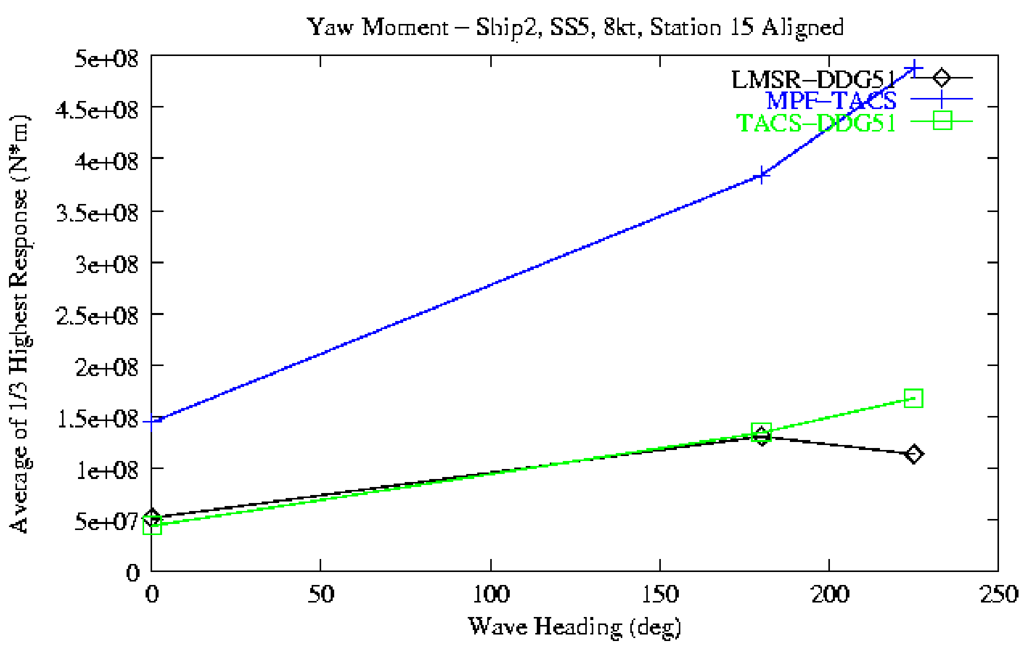

Figure 38 Yaw Moment with Station 15 Aligned in Sea State 5 at 8 knots for Ship 2

Validation Cases - Regular Waves, Zero Speed

As described in the Approach, above, the validation cases were first a set of irregular seas cases in the time domain. For direct comparison with the frequency domain, we performed regular wave cases with the waves shown in **Error! Reference source not found.**. These results are thus dimensional, and each wave has a different amplitude. These can be viewed as "Transfer Functions" for the specific heading, speed and sea state analyzed. Note that for waves close to the modal period of the spectrum, the frequency density is higher and the wave amplitudes may be smaller, as the wave amplitude depends on spectral density and frequency density.

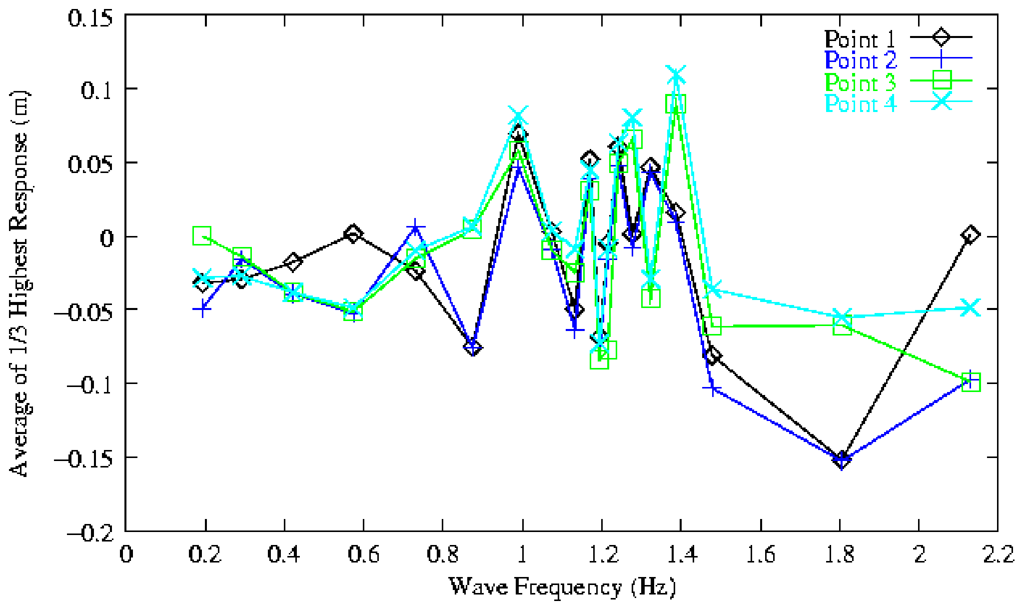

Figure 27 Relative Distance vs. Frequency in Sea State 3 (composed of individual Regular Waves) at 0 Degree Heading (LMSR-TACS)

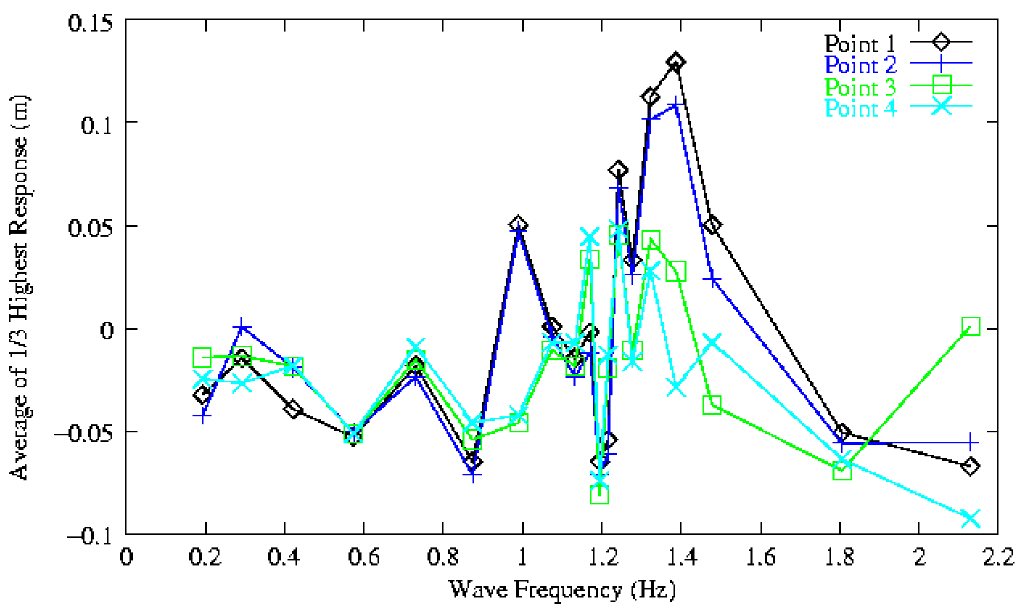

Figure 28 Relative Distance vs. Frequency in Sea State 3 (composed of individual Regular Waves) at 180 Degree Heading (LMSR-TACS)

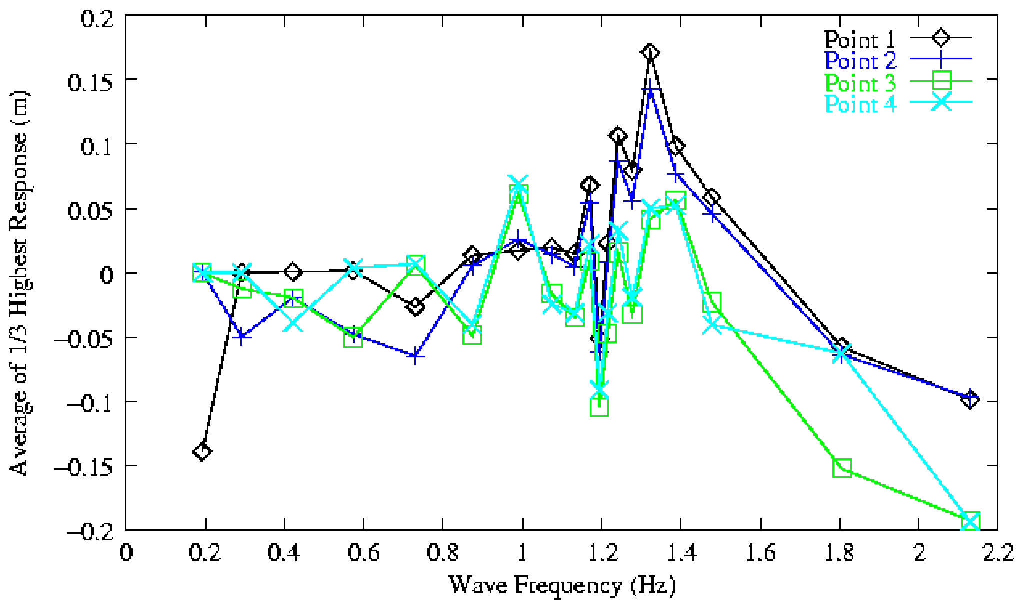

Figure 29 Relative Distance vs. Frequency in Sea State 3 (composed of individual Regular Waves) at 225 Degree Heading (LMSR-TACS)

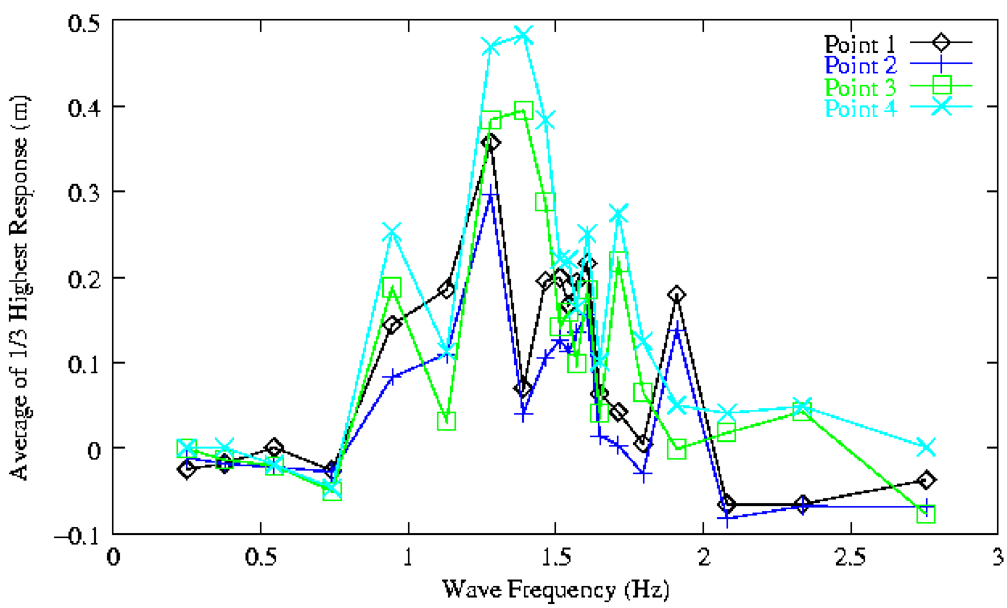

Figure 30 Relative Distance vs. Frequency in Sea State 5 (composed of individual Regular Waves) at 0 Degree Heading (LMSR-TACS)

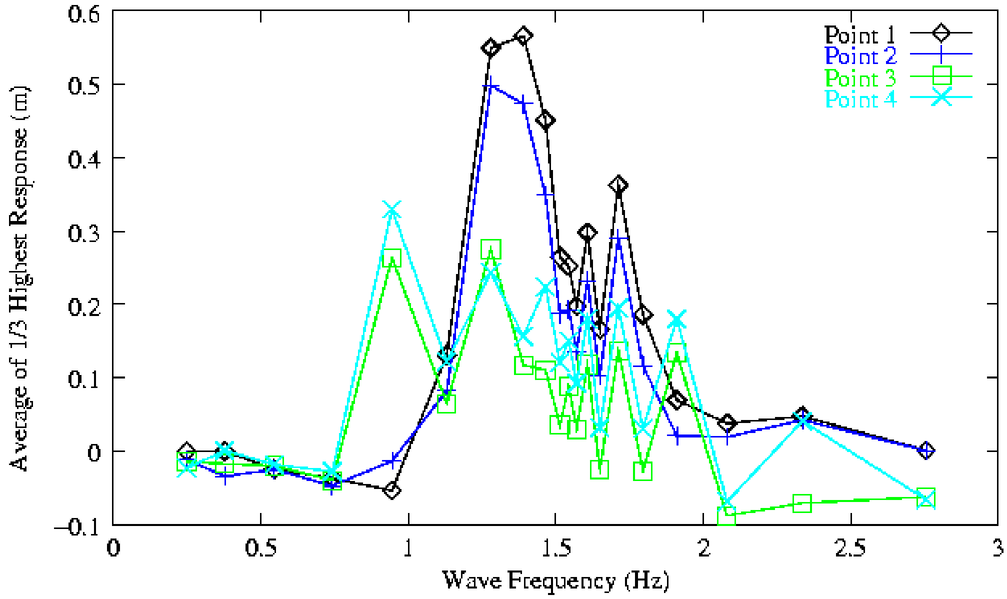

Figure 31 Relative Distance vs. Frequency in Sea State 5 (composed of individual Regular Waves) at 180 Degree Heading (LMSR-TACS)

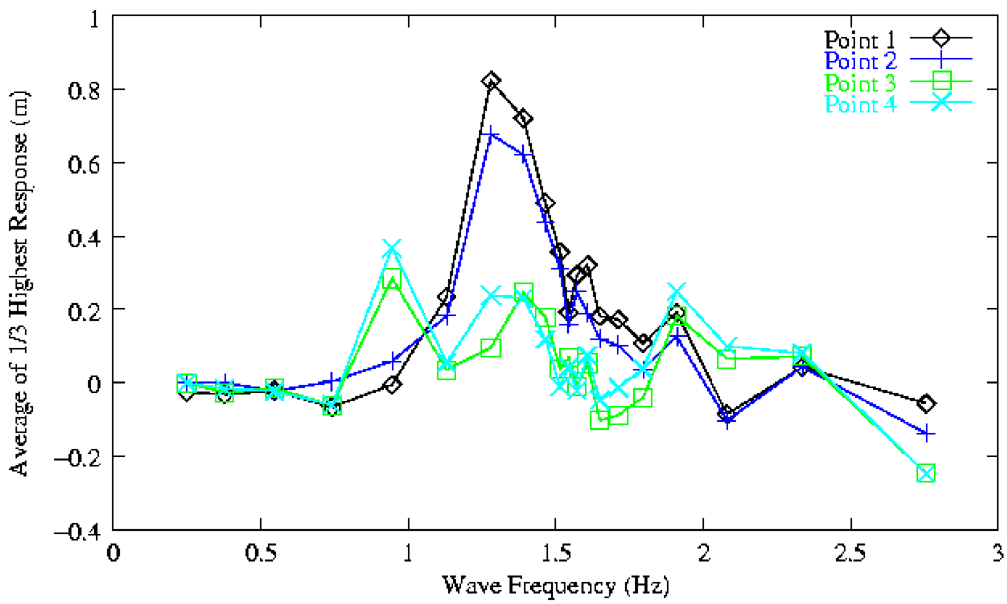

Figure 32 Relative Distance vs. Frequency in Sea State 5 (composed of individual Regular Waves) at 225 Degree Heading (LMSR-TACS)

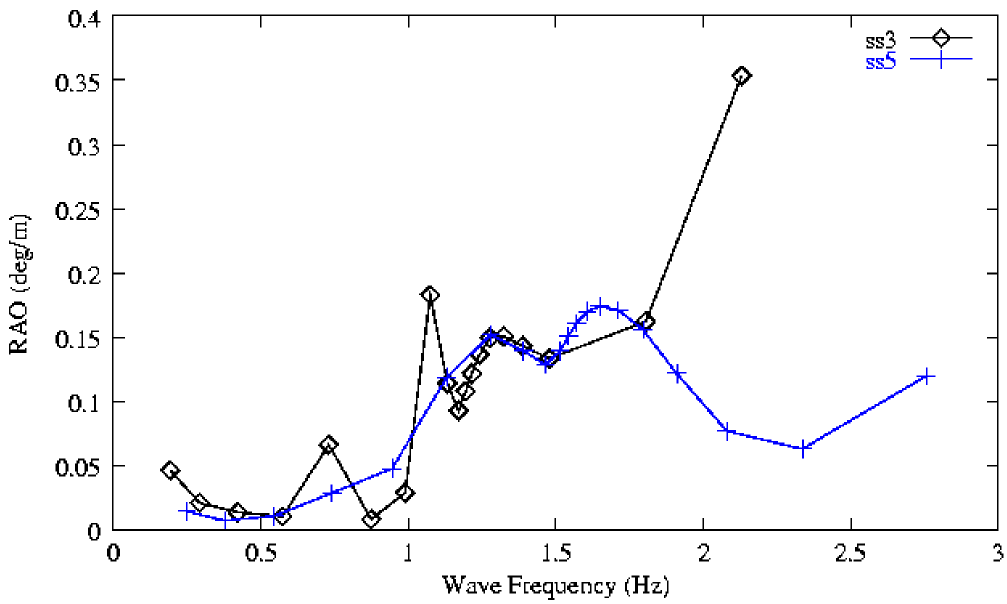

Figure 33 LMSR Roll Angle vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

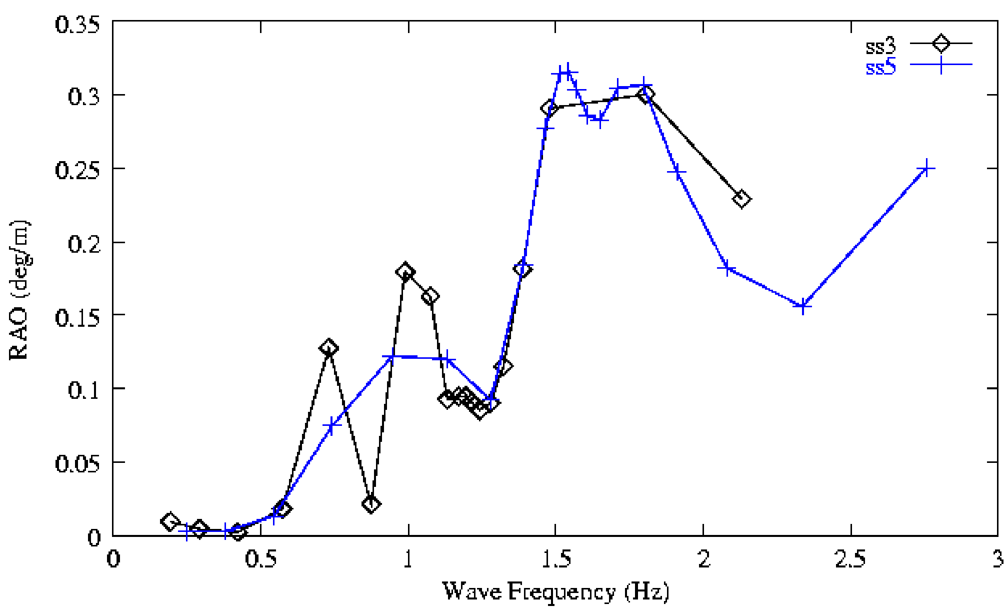

Figure 34 TACS Roll Angle vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

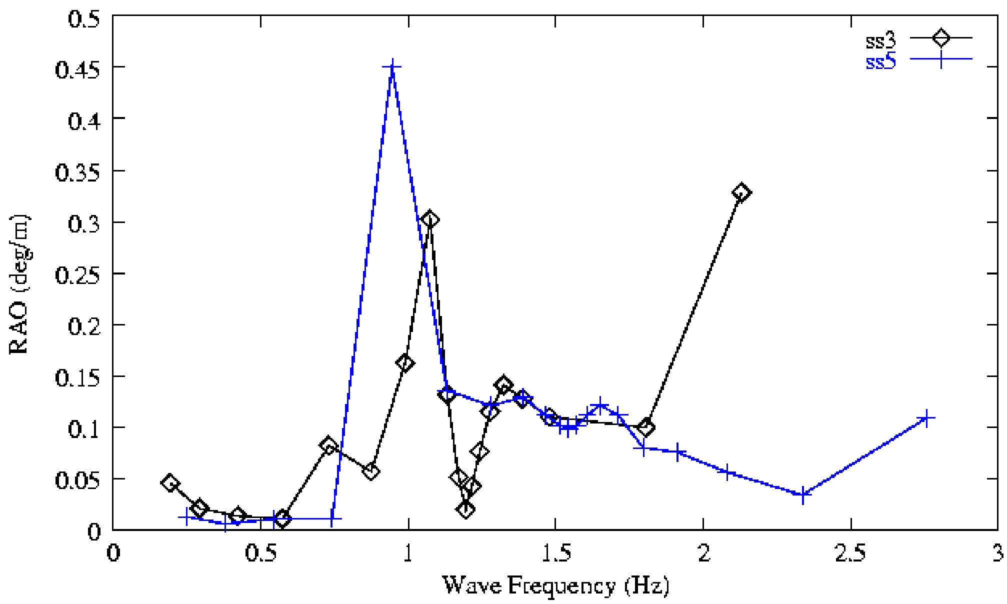

Figure 35 LMSR Roll Angle vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

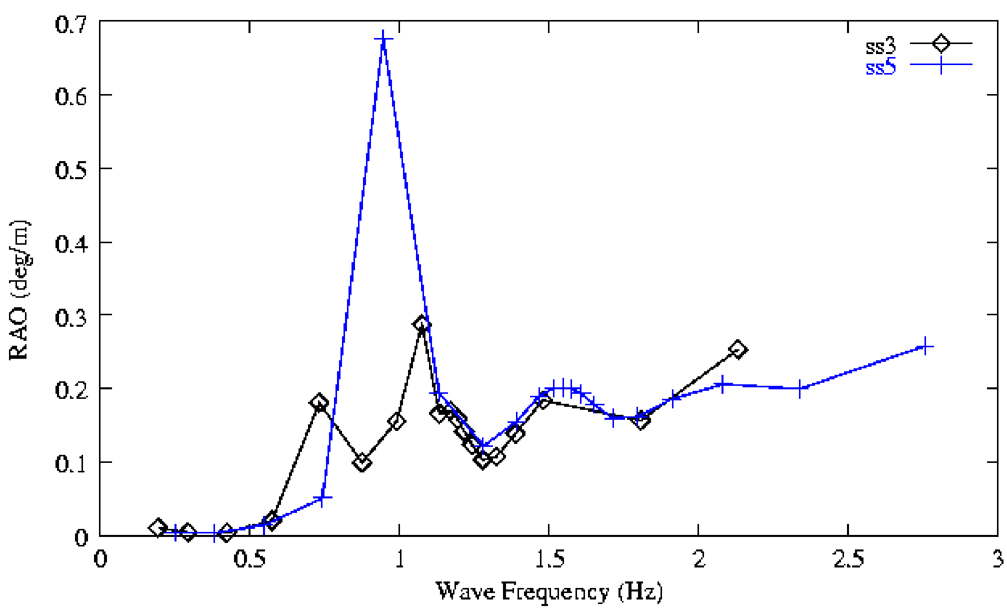

Figure 36 TACS Roll Angle vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

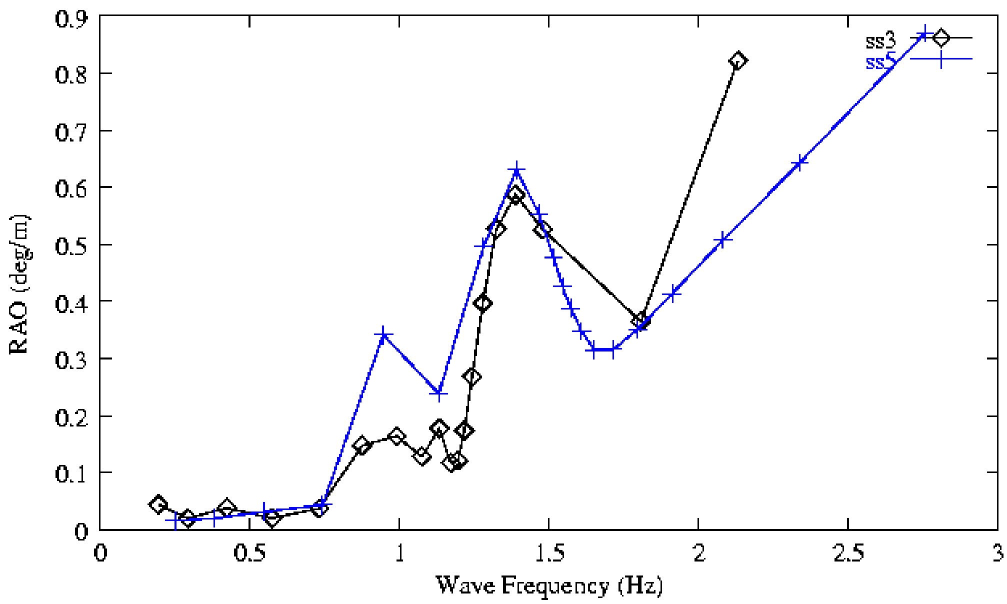

Figure 37 LMSR Roll Angle vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

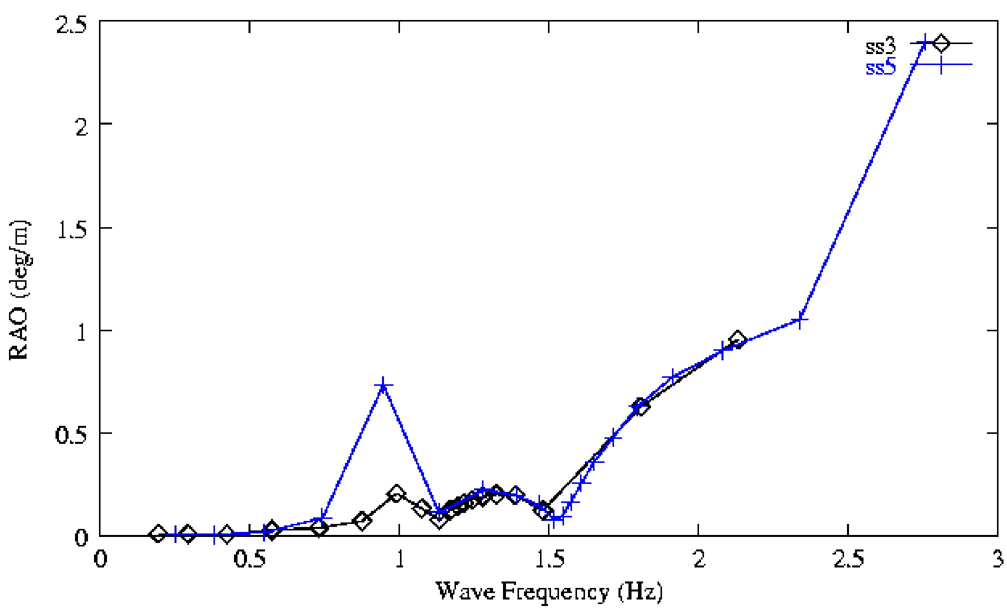

Figure 38 TACS Roll Angle vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

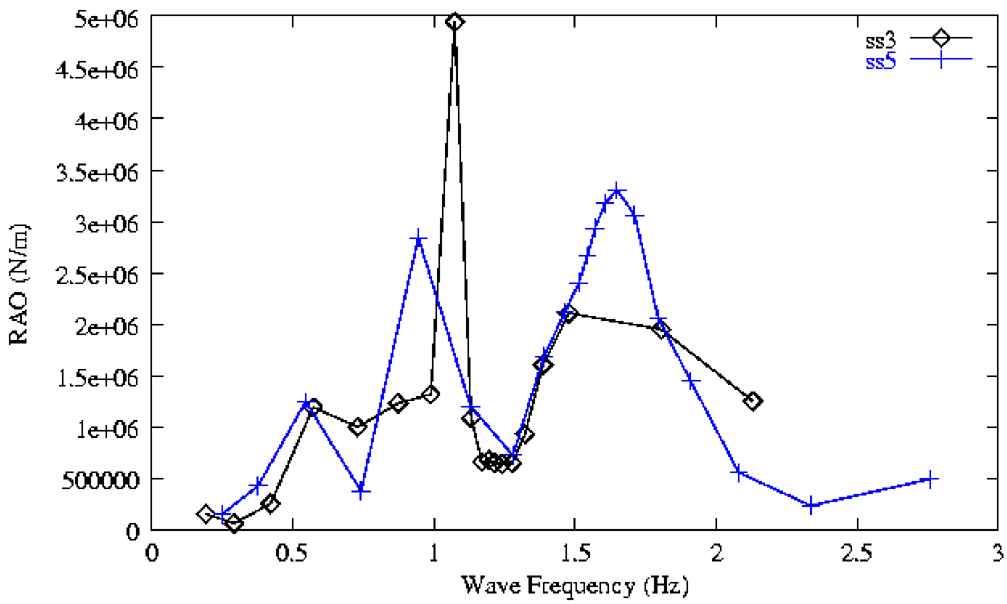

Figure 39 LMSR Sway Force vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

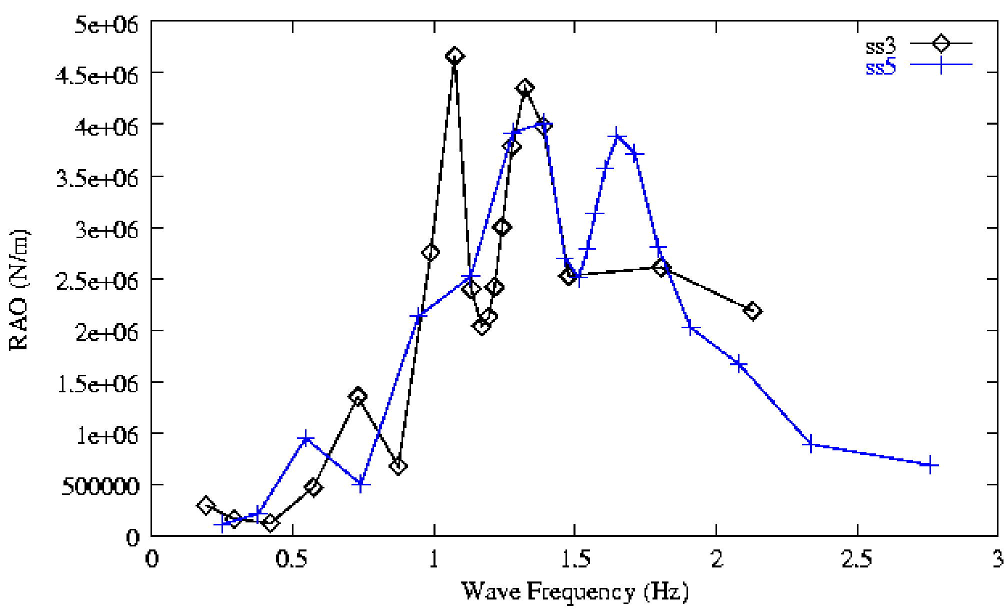

Figure 40 TACS Sway Force vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

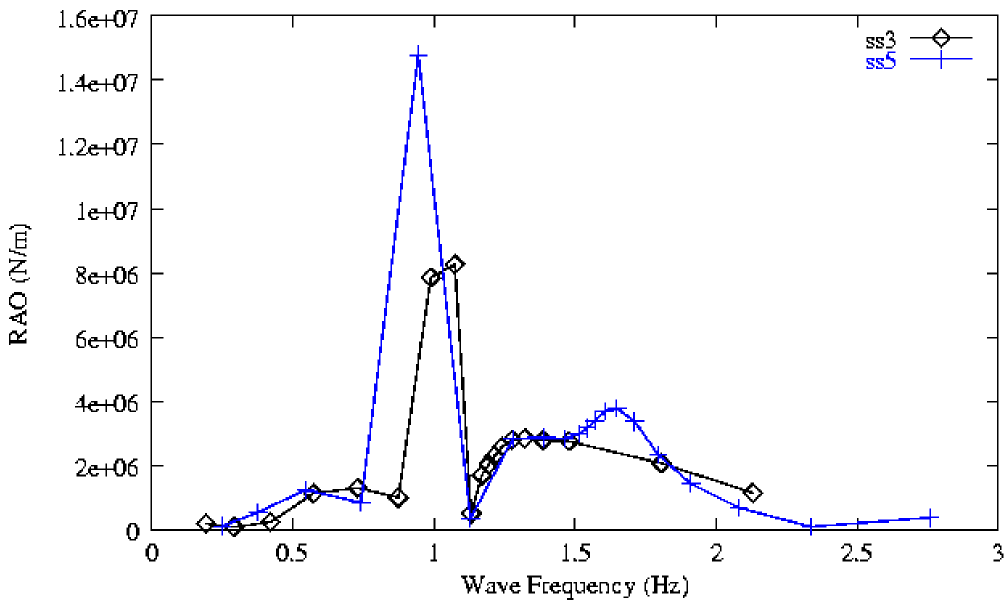

Figure 41 LMSR Sway Force vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

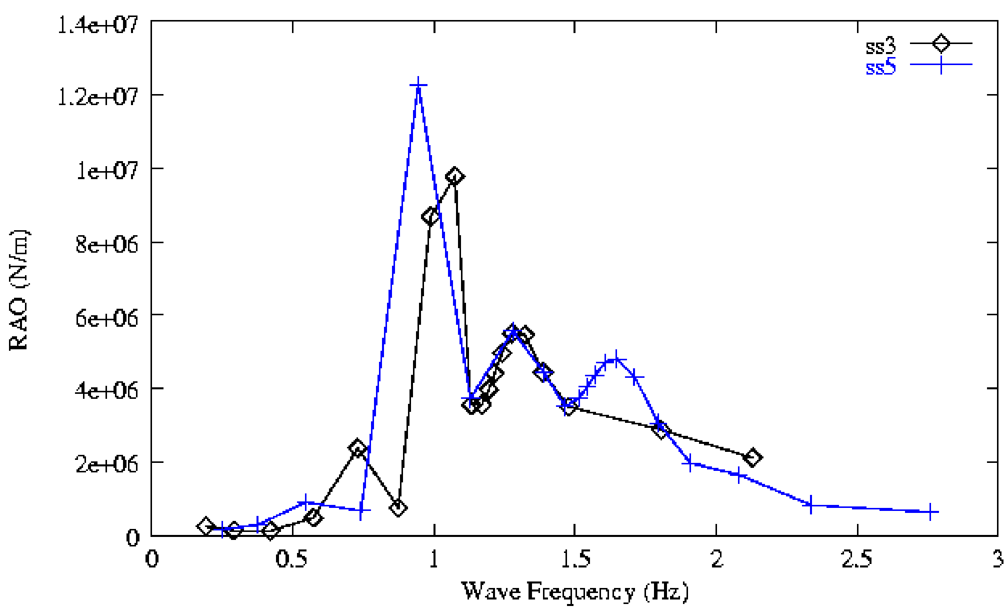

Figure 42 TACS Sway Force vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

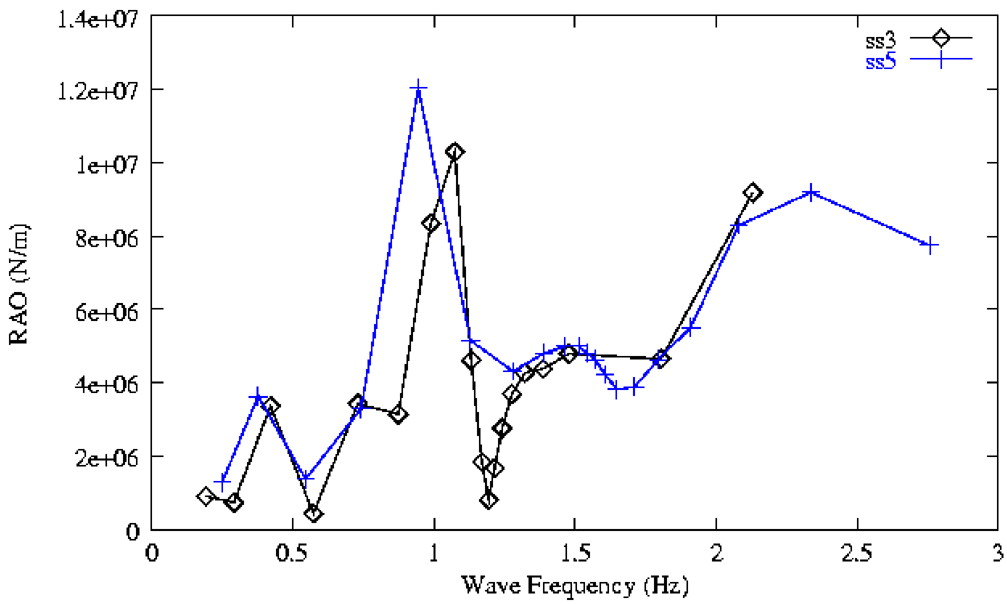

Figure 43 LMSR Sway Force vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

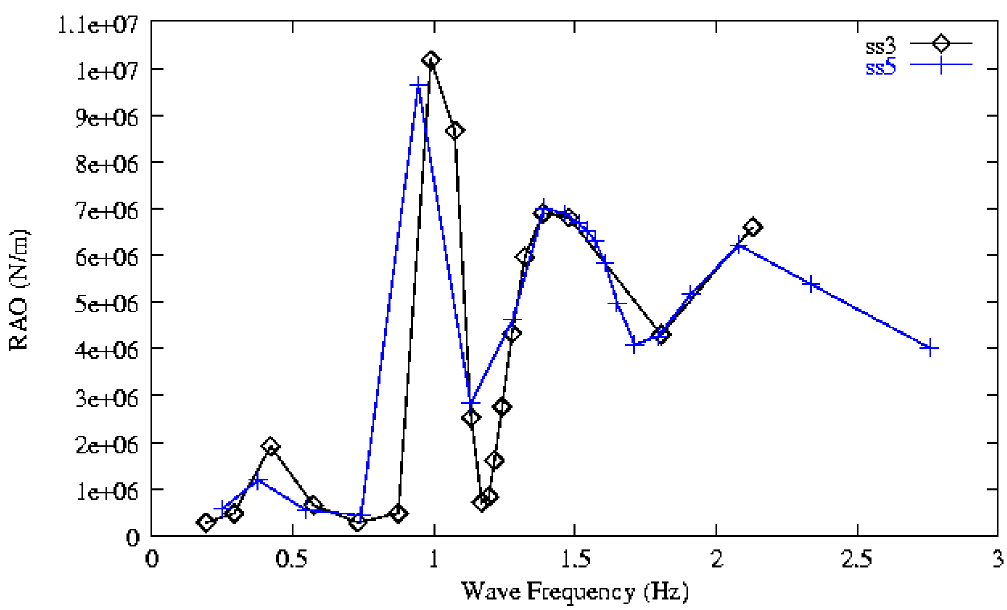

Figure 44 TACS Sway Force vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

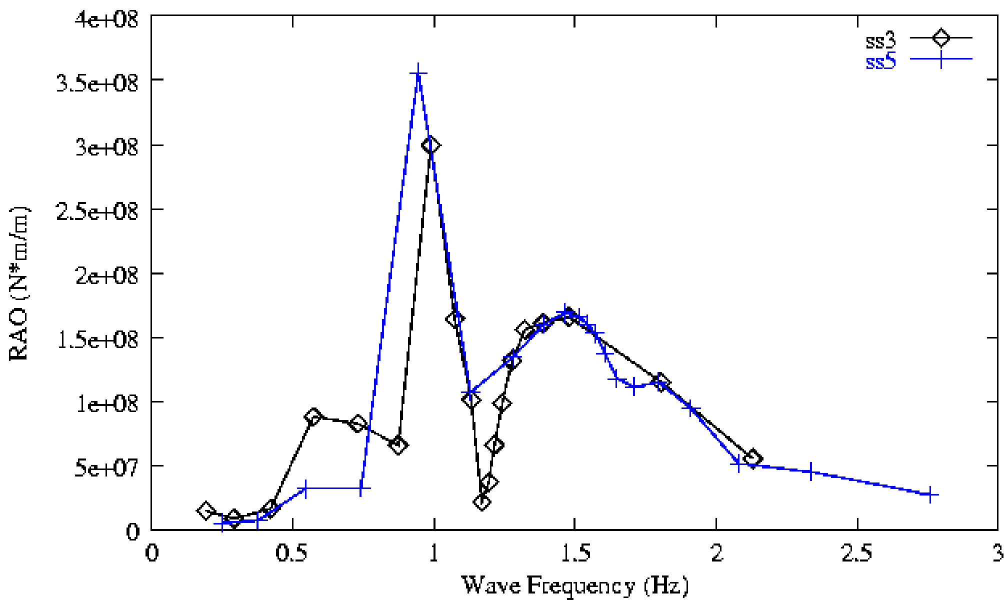

Figure 45 LMSR Yaw Moment vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

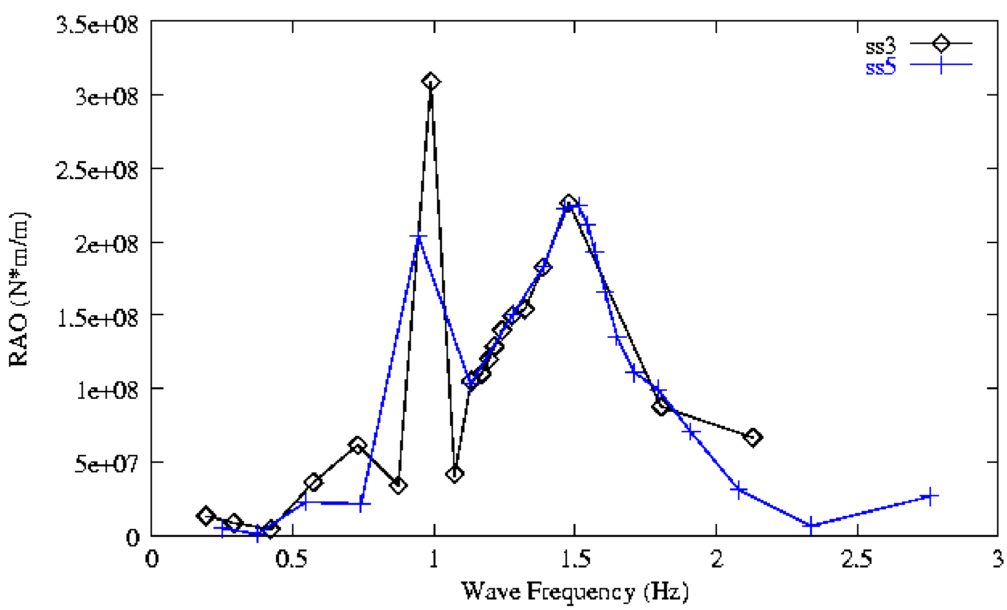

Figure 46 TACS Yaw Moment vs. Wave Freq. at 0 Degree Heading (LMSR-TACS)

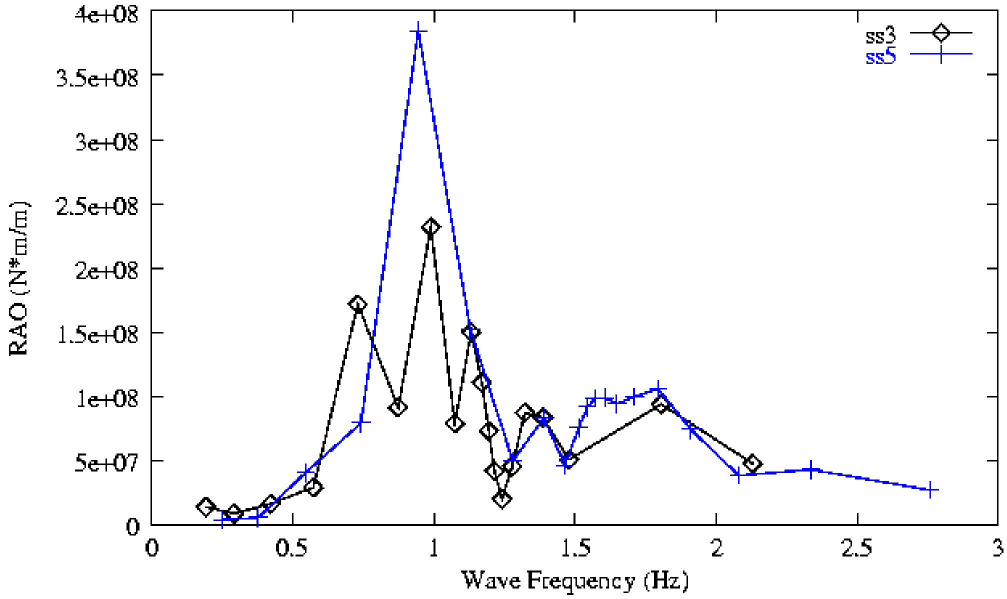

Figure 47 LMSR Yaw Moment vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

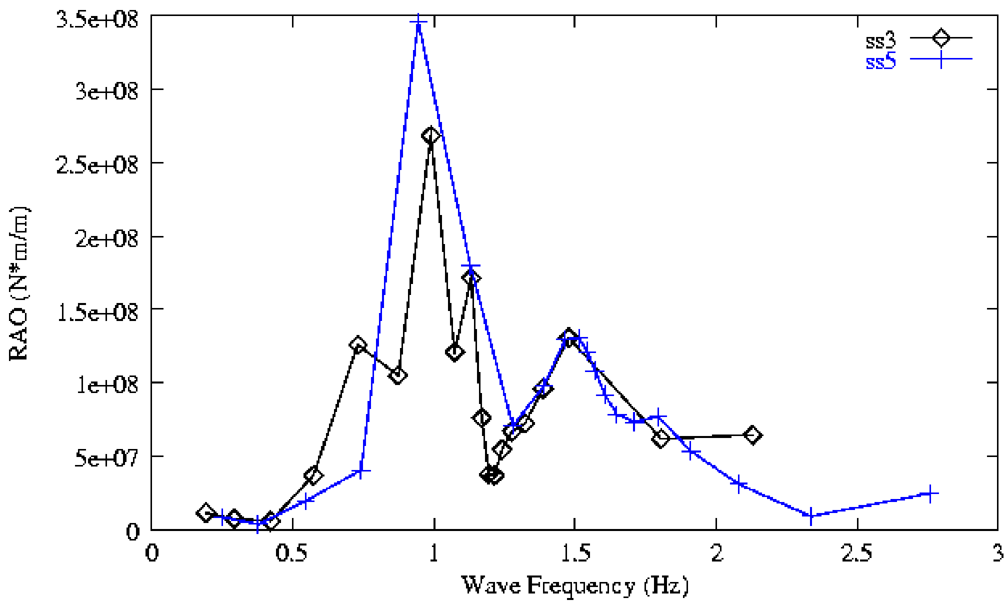

Figure 48 TACS Yaw Moment vs. Wave Freq. at 180 Degree Heading (LMSR-TACS)

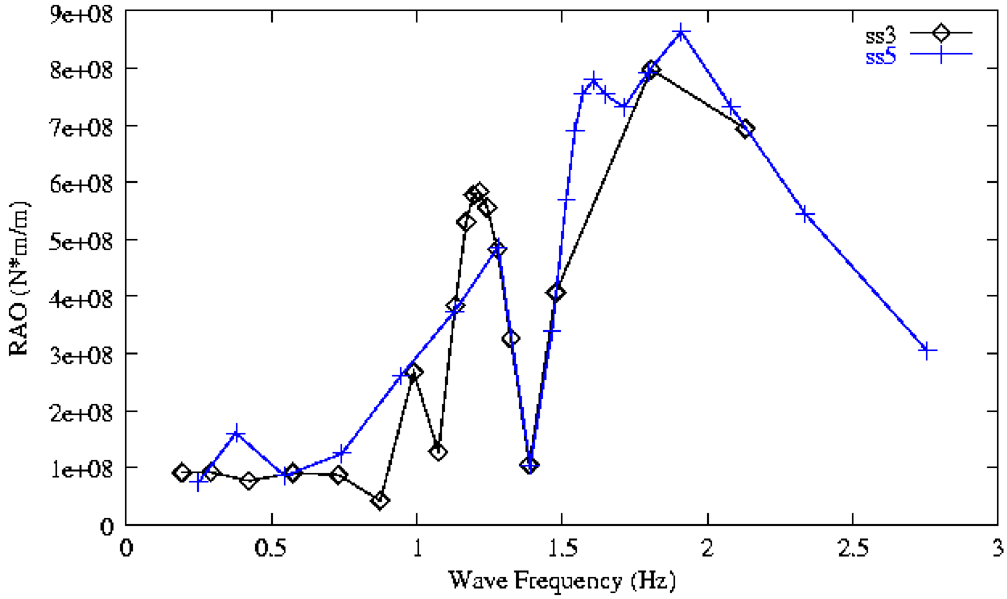

Figure 49 LMSR Yaw Moment vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

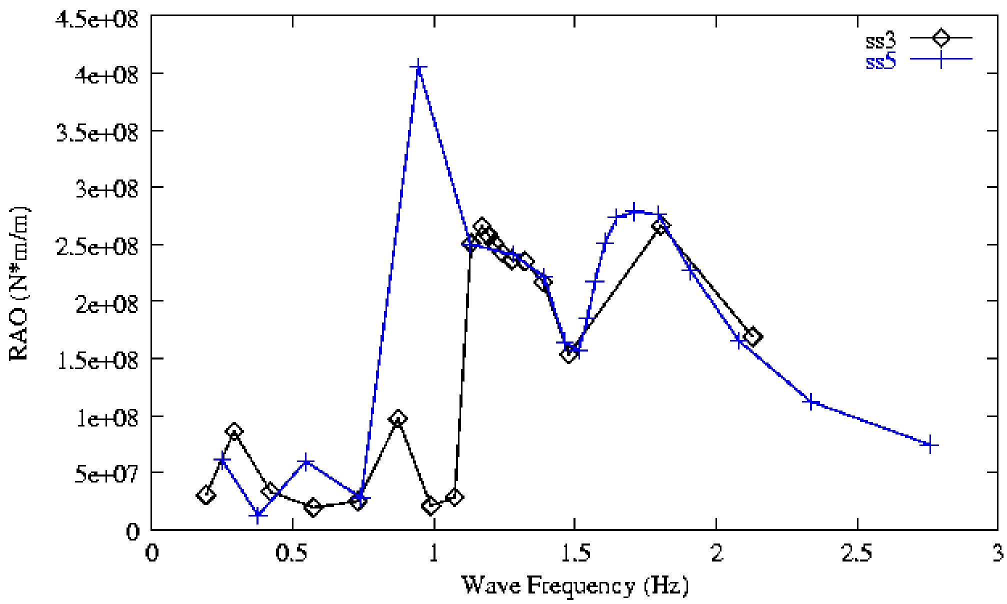

Figure 50 TACS Yaw Moment vs. Wave Freq. at 225 Degree Heading (LMSR-TACS)

Regular Wave Cases

Figure 51 through Figure 60 show regular wave relative motions as dimensional response, with no normalization. Figure 61 through Figure 78 give the regular wave motions response as RAO's, with all responses divided by wave amplitude.

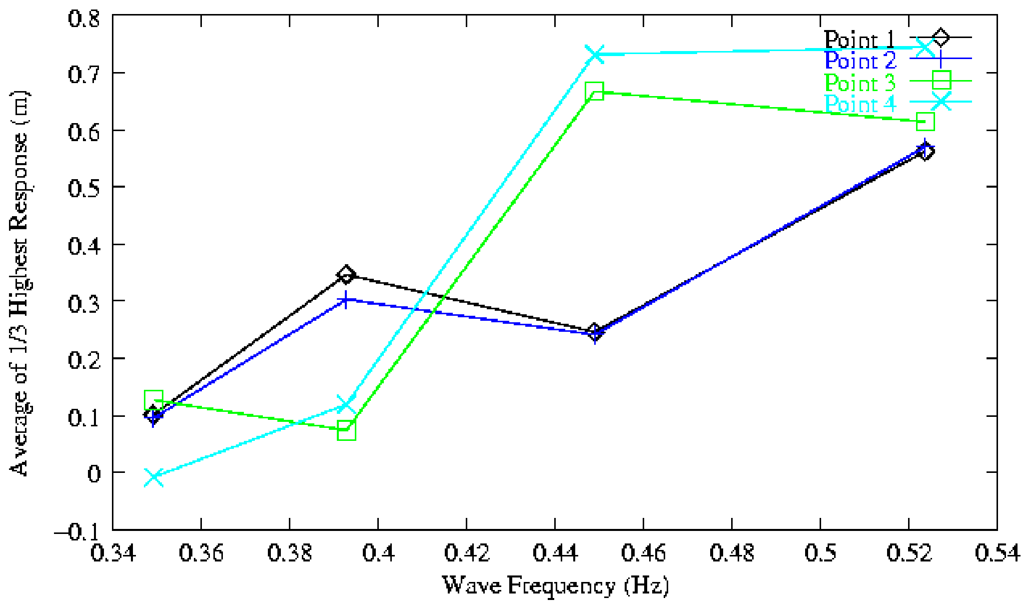

Figure 51 Relative Distance vs. Wave Freq. for 1.5m Wave Amplitude 0 degree Heading

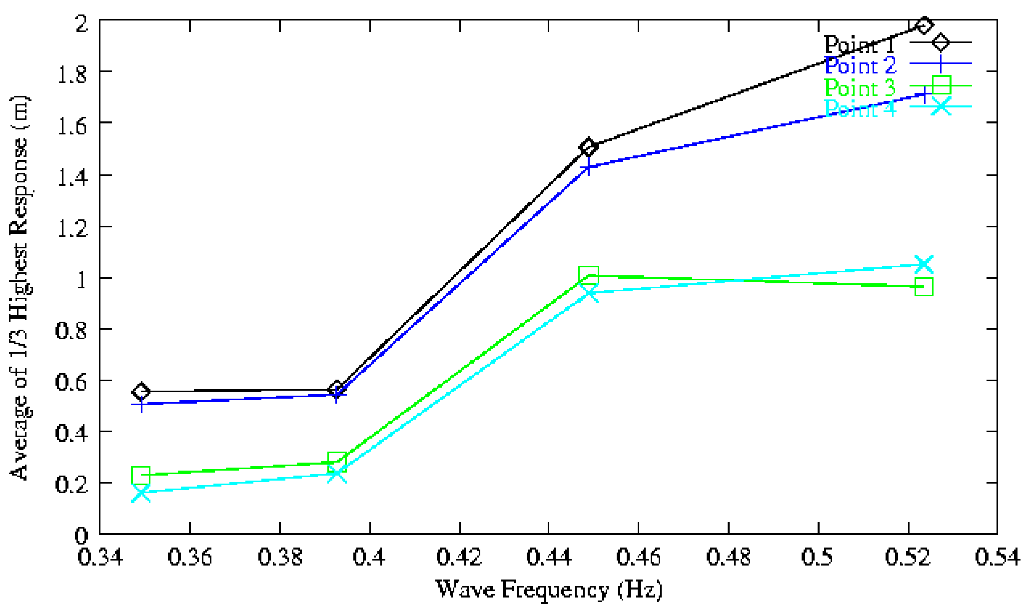

Figure 52 Relative Distance vs. Wave Freq. for 1.5m Wave Amplitude 180 degree Heading

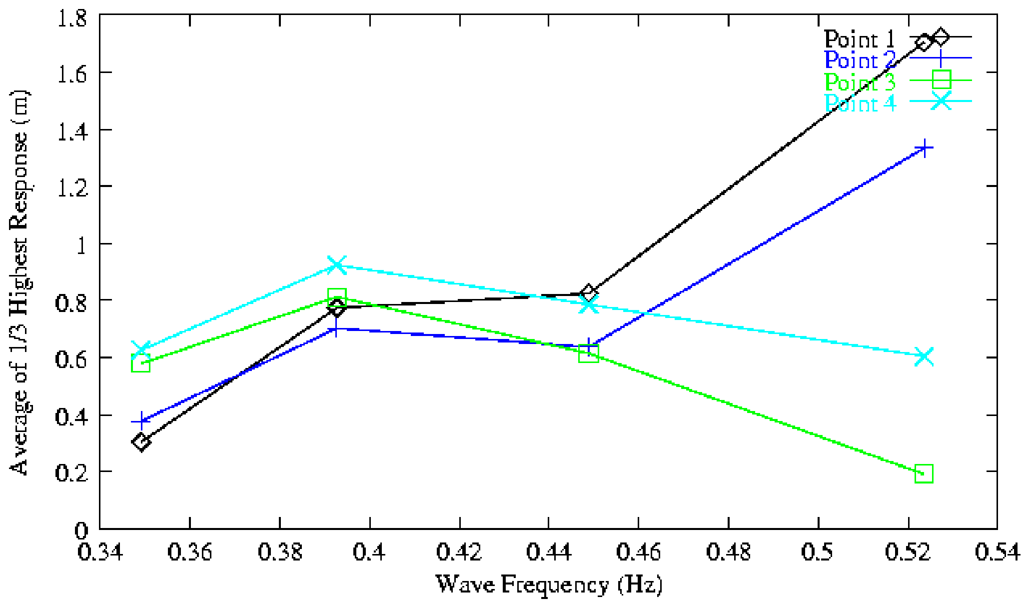

Figure 53 Relative Distance vs. Wave Freq. for 1.5m Wave Amplitude 225 degree Heading

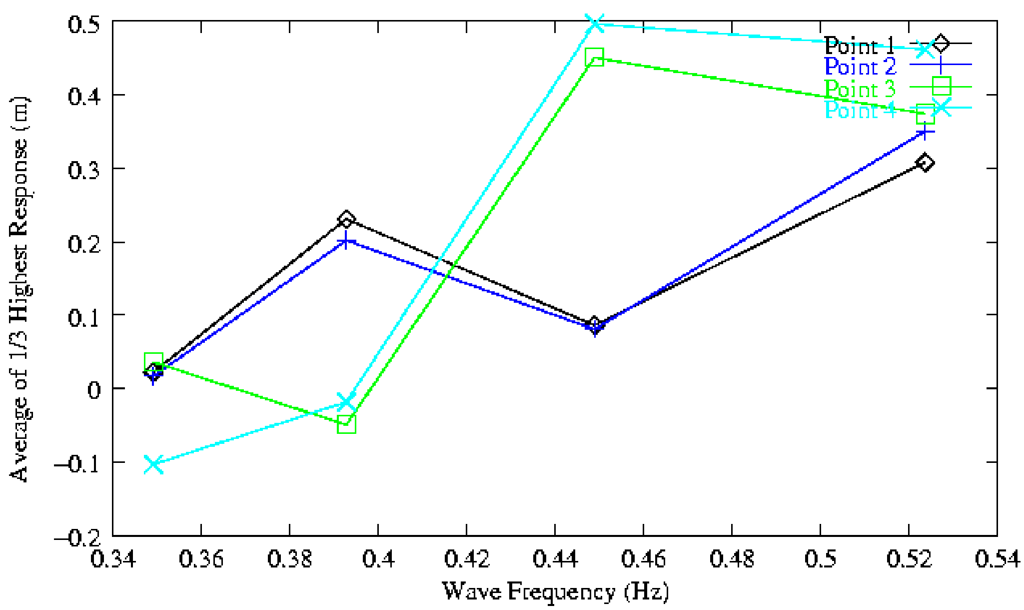

Figure 54 Relative Distance vs. Wave Freq. for 1m Wave Amplitude 0 degree Heading

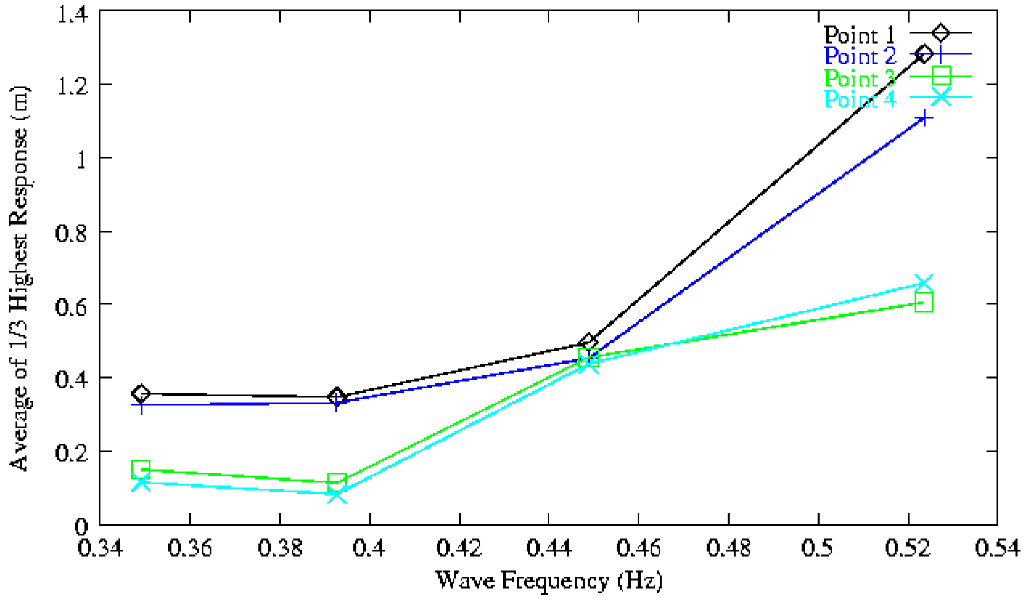

Figure 55 Relative Distance vs. Wave Freq. for 1m Wave Amplitude 180 degree Heading

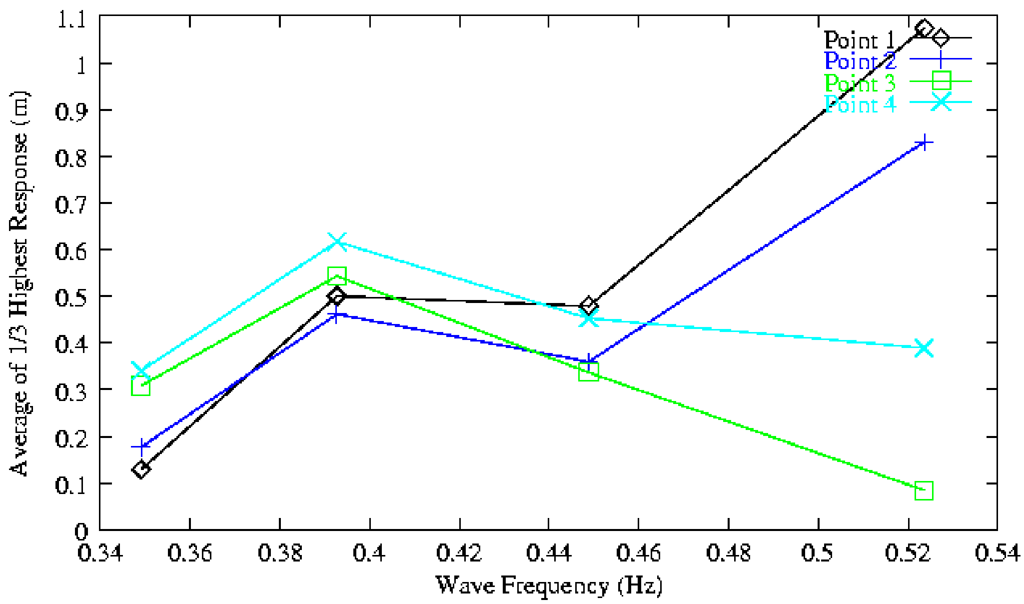

Figure 56 Relative Distance vs. Wave Freq. for 1m Wave Amplitude 225 degree Heading

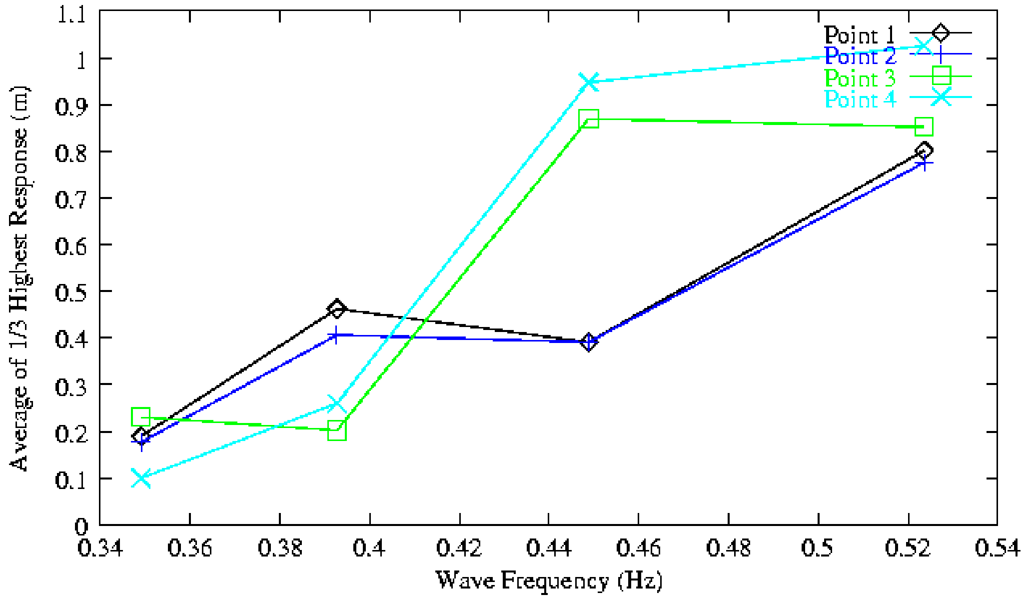

Figure 57 Relative Distance vs. Wave Freq. for 2m Wave Amplitude 0 degree Heading

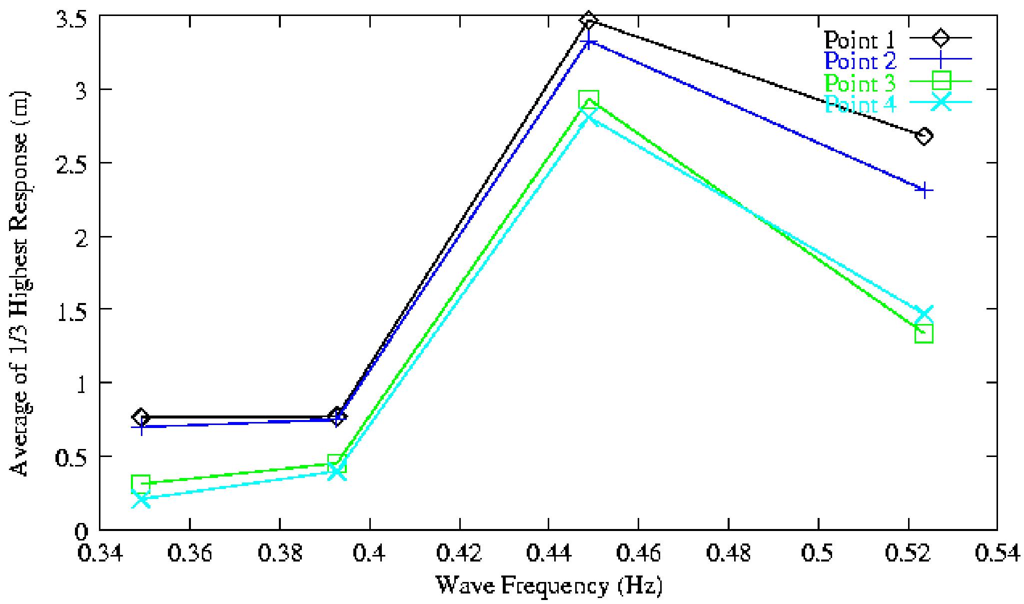

Figure 58 Relative Distance vs. Wave Freq. for 2m Wave Amplitude 180 degree Heading

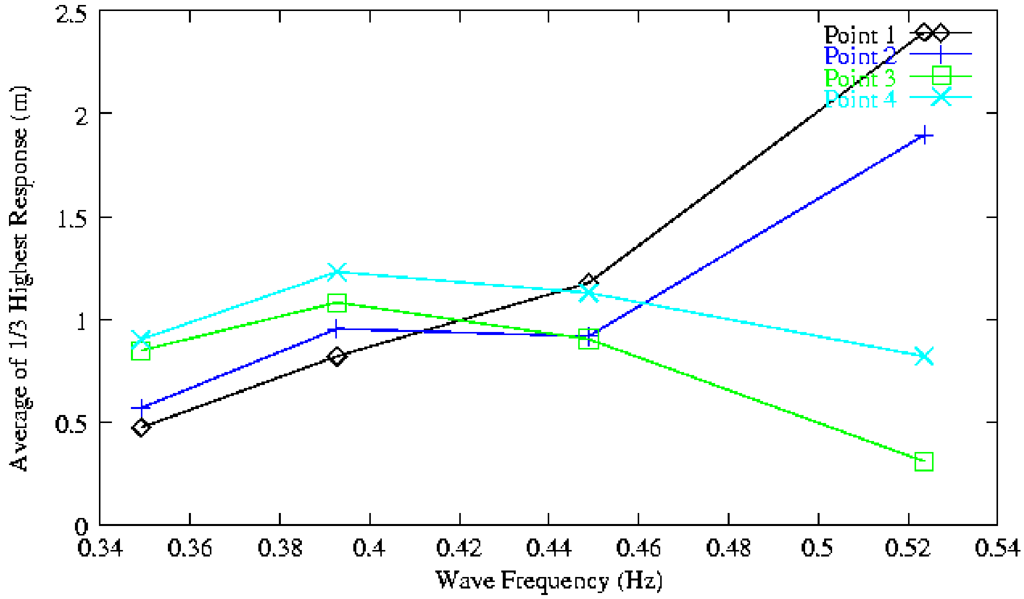

Figure 59 Relative Distance vs. Wave Freq. for 2m Wave Amplitude 225 degree Heading

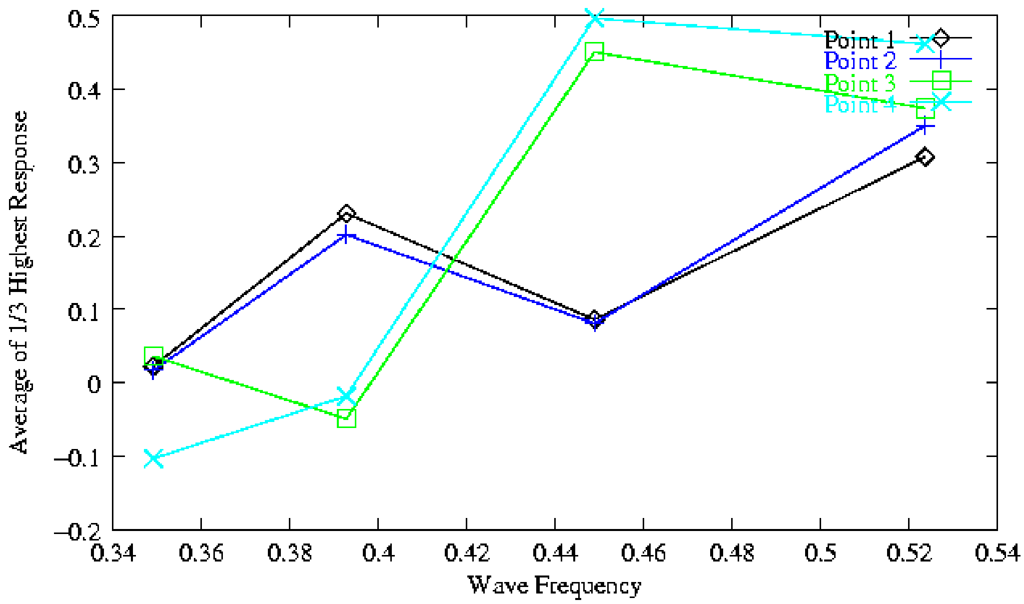

Figure 60 Relative Distance vs. Wave Freq. for 1m Wave Amplitude 0 Degree Heading

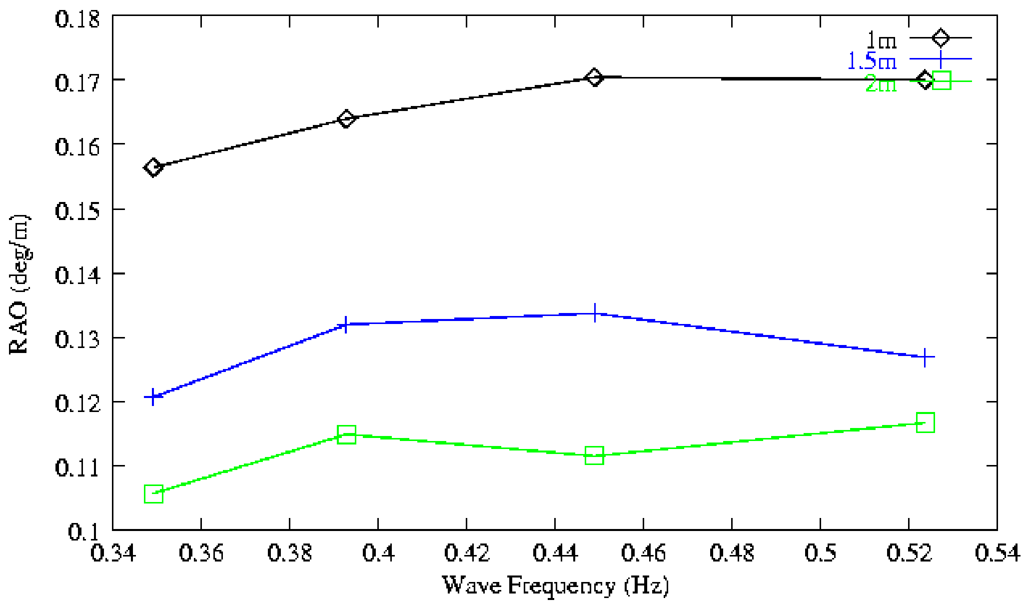

Figure 61 LMSR Roll Angle vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

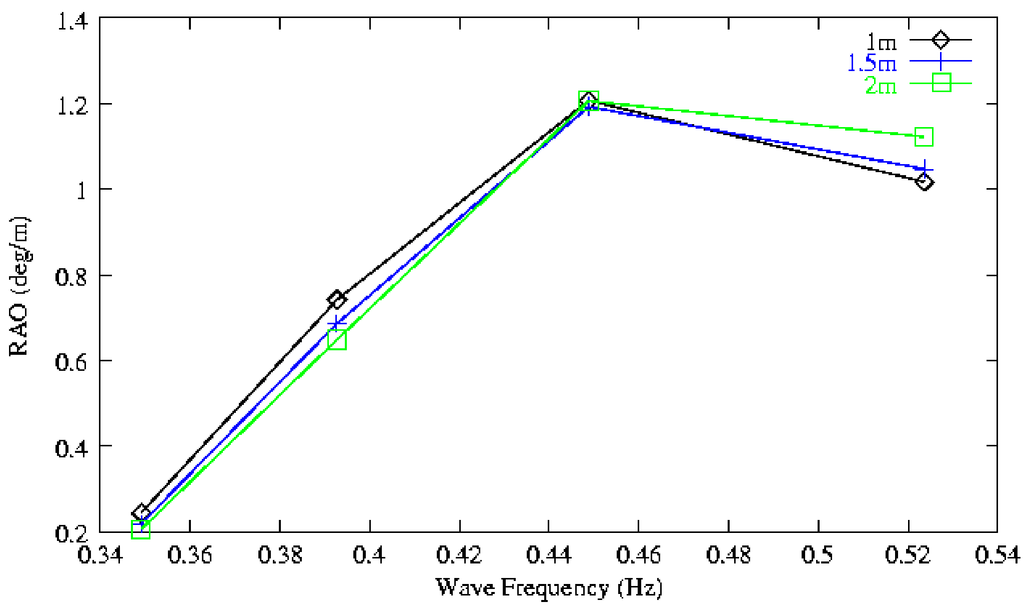

Figure 62 TACS Roll Angle vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

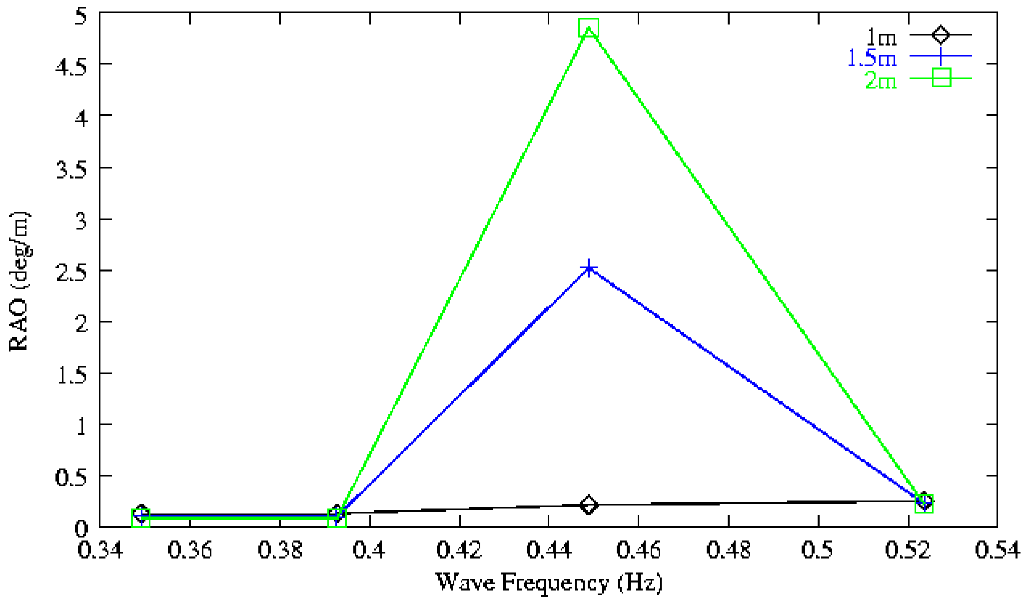

Figure 63 LMSR Roll Angle vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

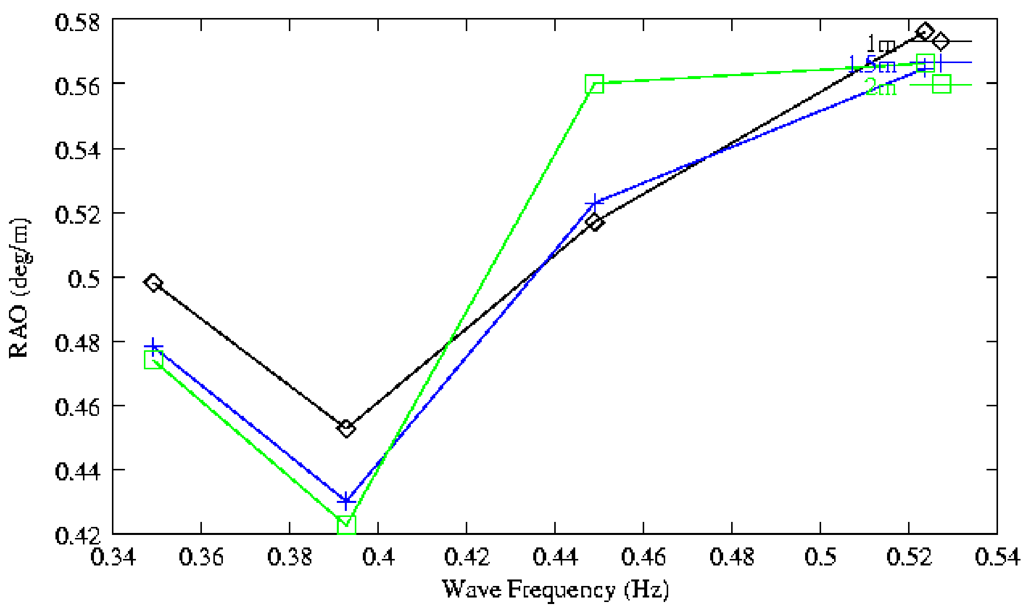

Figure 64 TACS Roll Angle vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

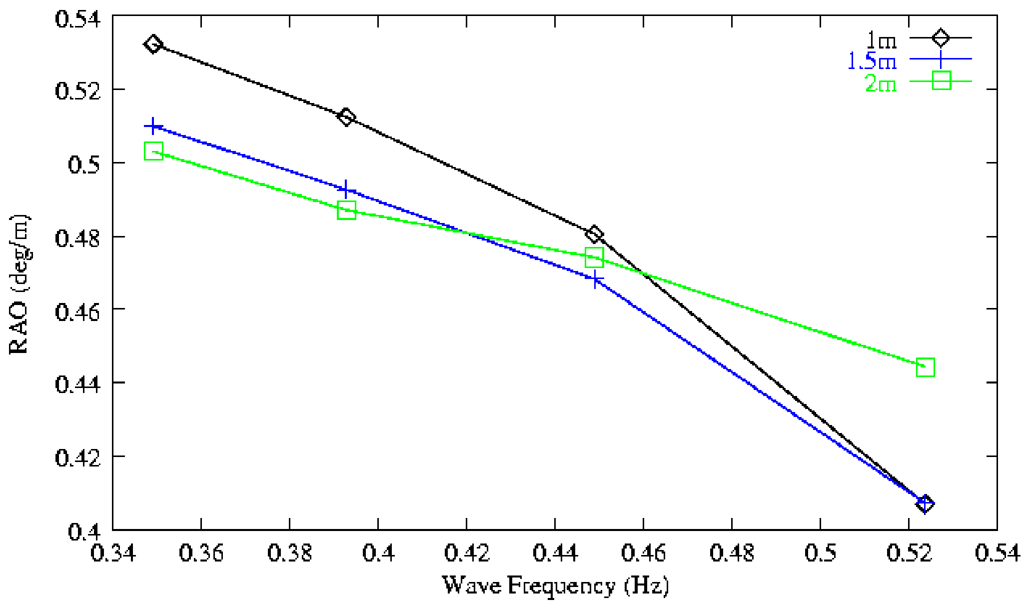

Figure 65 LMSR Roll Angle vs. Wave Freq._225deg (LMSR-TACS)

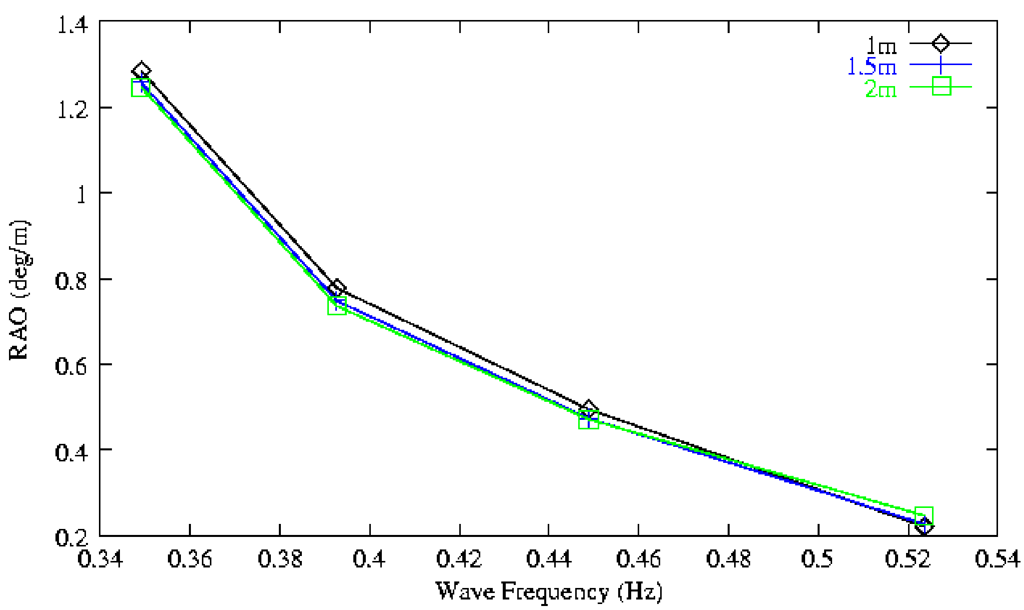

Figure 66 TACS Roll Angle vs. Wave Freq._225deg (LMSR-TACS)

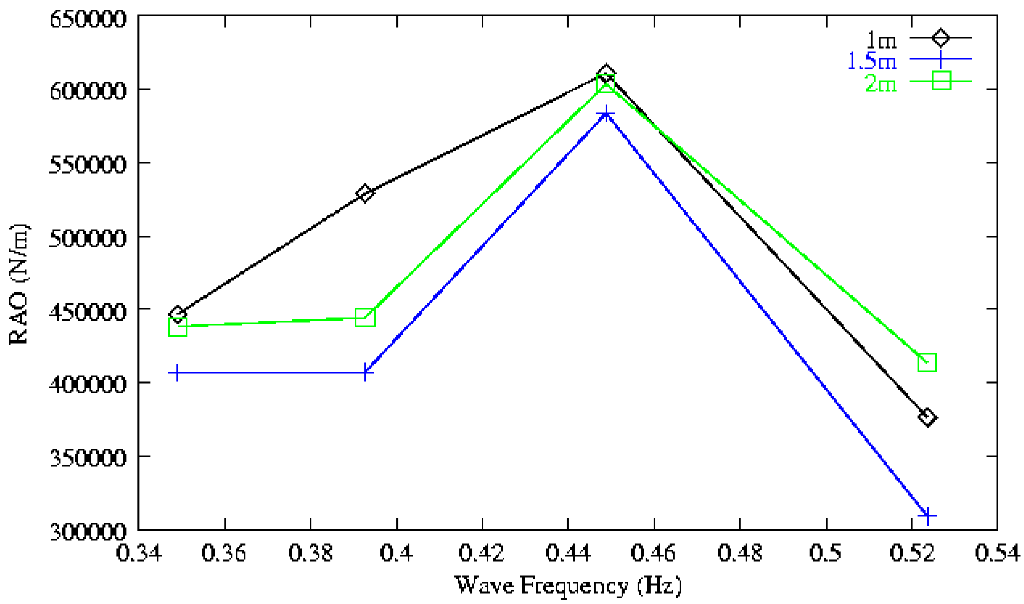

Figure 67 LMSR Sway Force vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

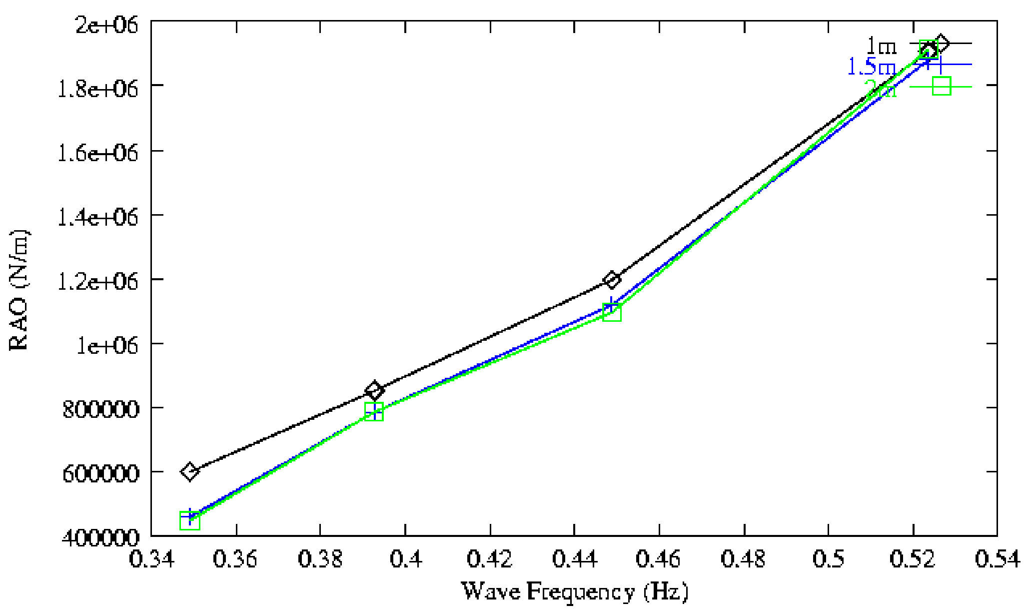

Figure 68 TACS Sway Force vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

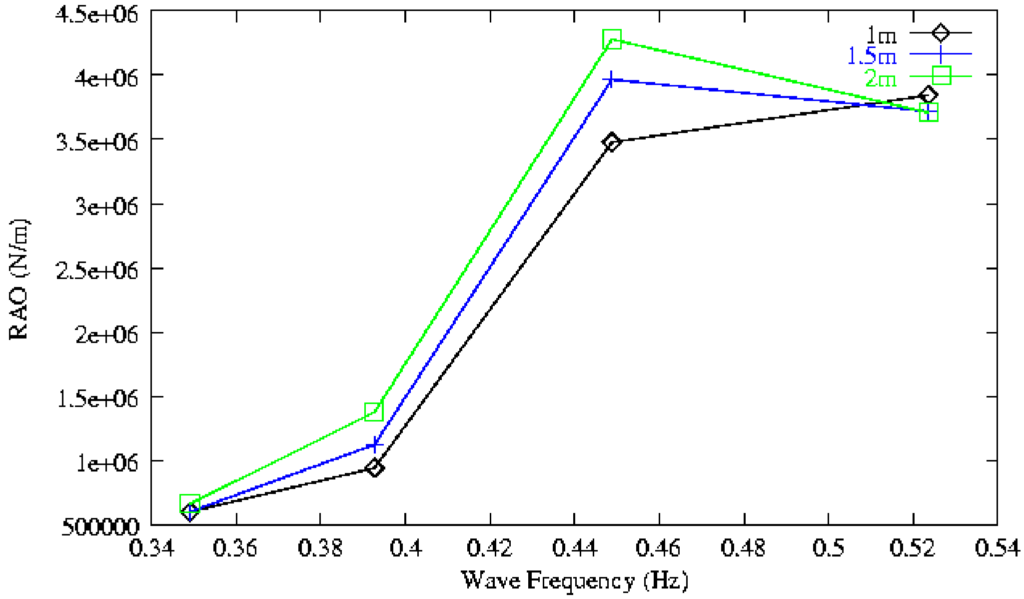

Figure 69 LMSR Sway Force vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

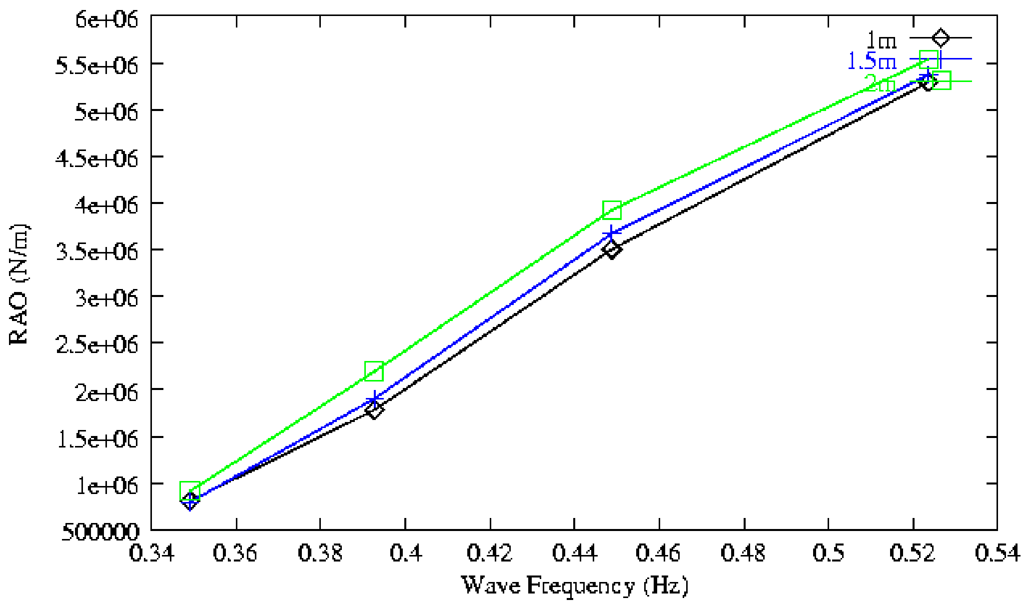

Figure 70 TACS Sway Force vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

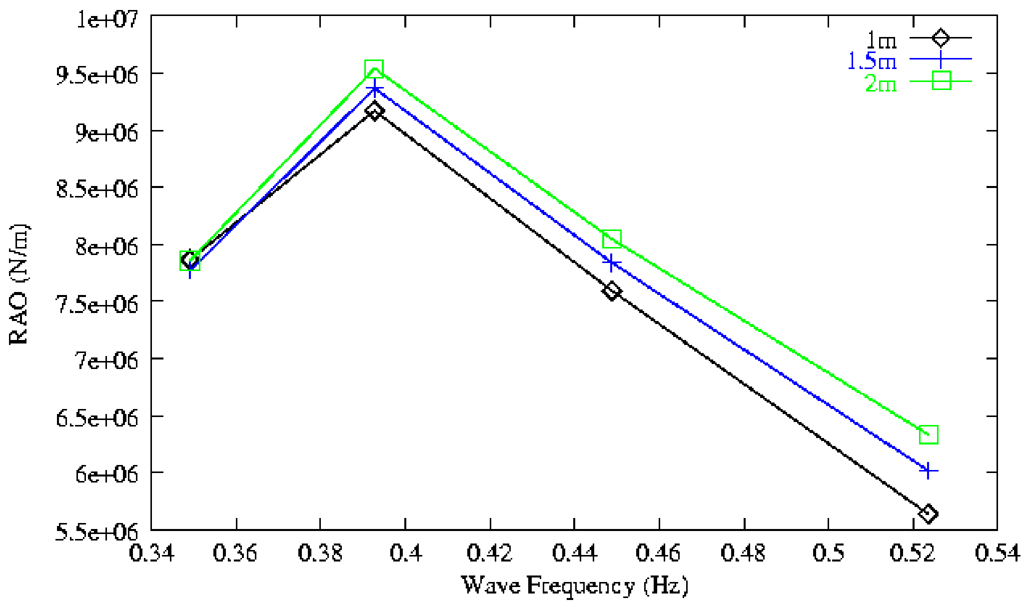

Figure 71 LMSR Sway Force vs. Wave Freq._225deg (LMSR-TACS)

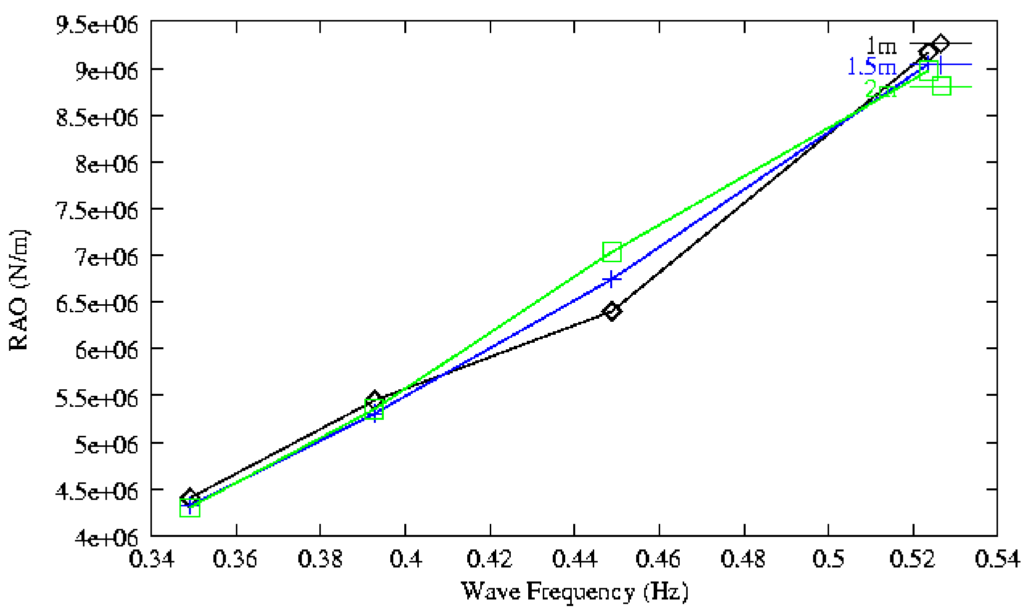

Figure 72 TACS Sway Force vs. Wave Freq._225deg (LMSR-TACS)

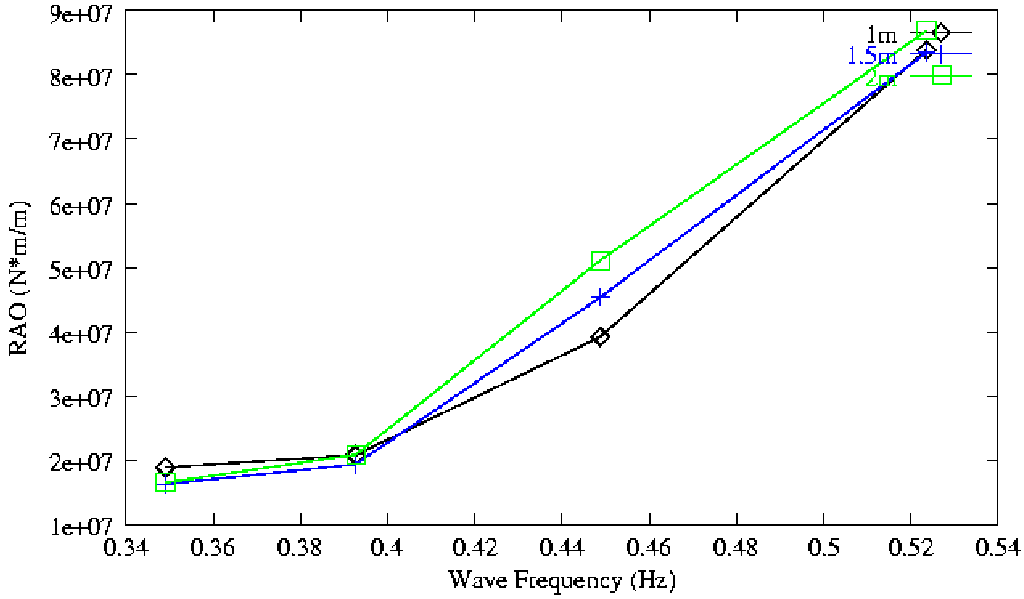

Figure 73 LMSR Yaw Moment vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

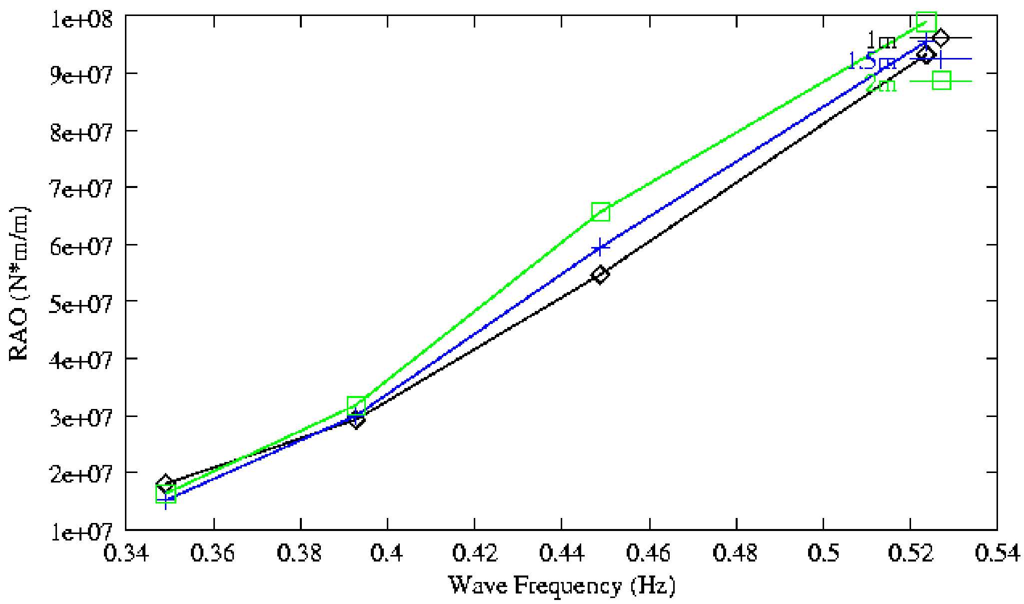

Figure 74 TACS Yaw Moment vs. Wave Freq. for 0 Degree Heading (LMSR-TACS)

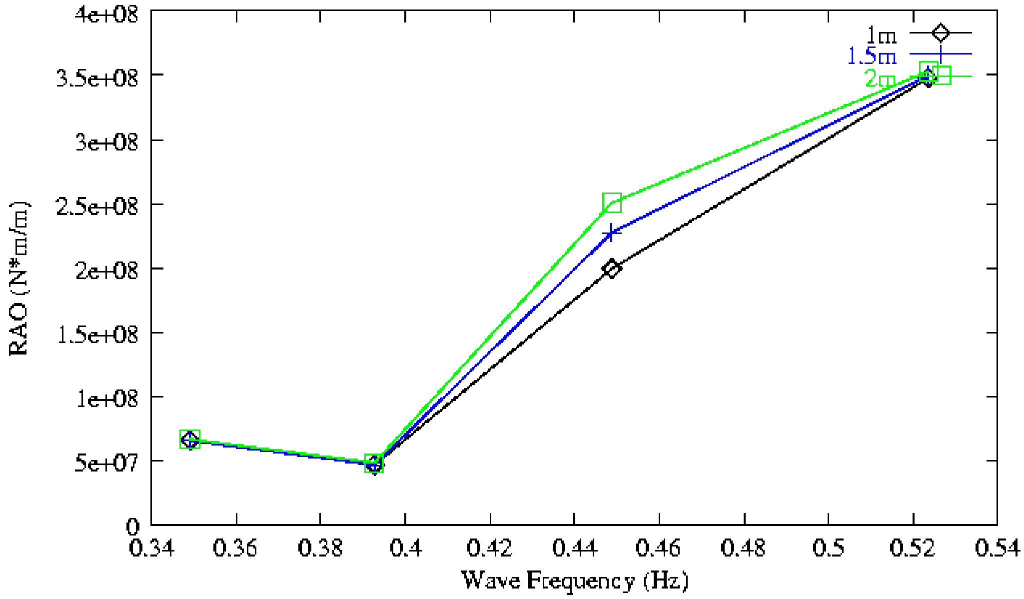

Figure 75 LMSR Yaw Moment vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

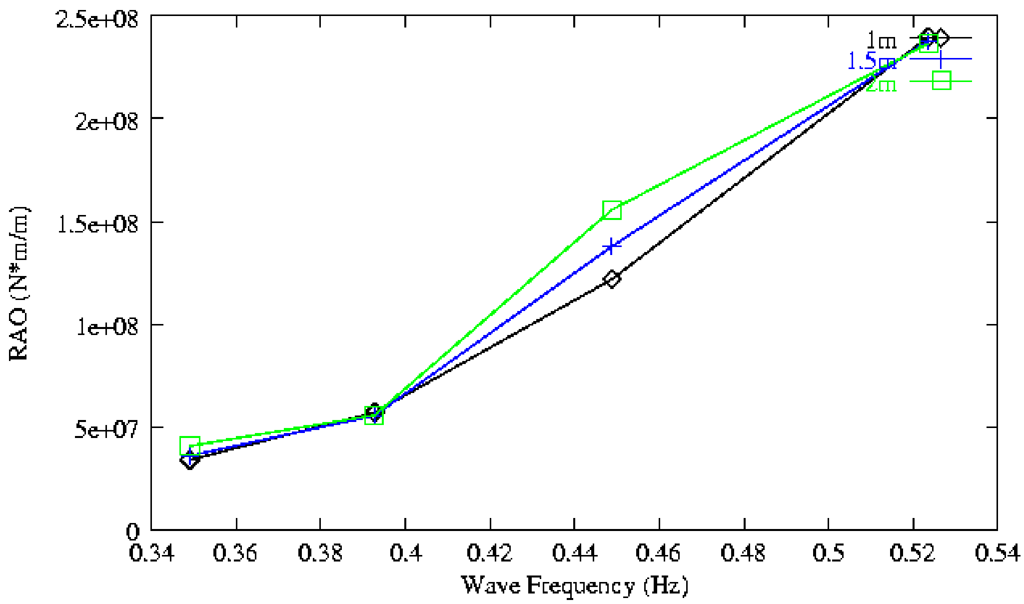

Figure 76 TACS Yaw Moment vs. Wave Freq. for 180 Degree Heading (LMSR-TACS)

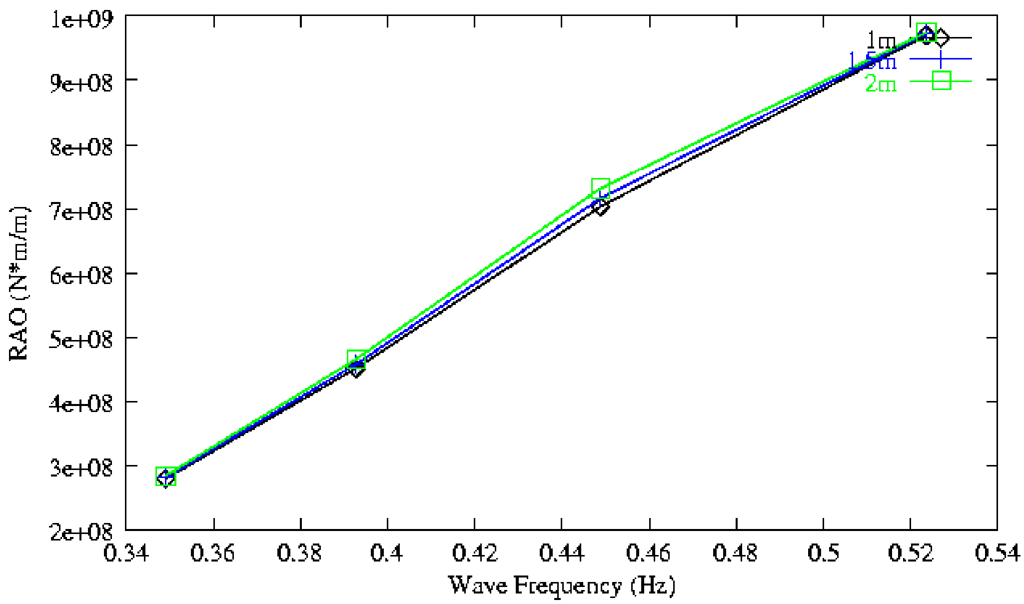

Figure 77 LMSR Yaw Moment vs. Wave Freq._225deg (LMSR-TACS)

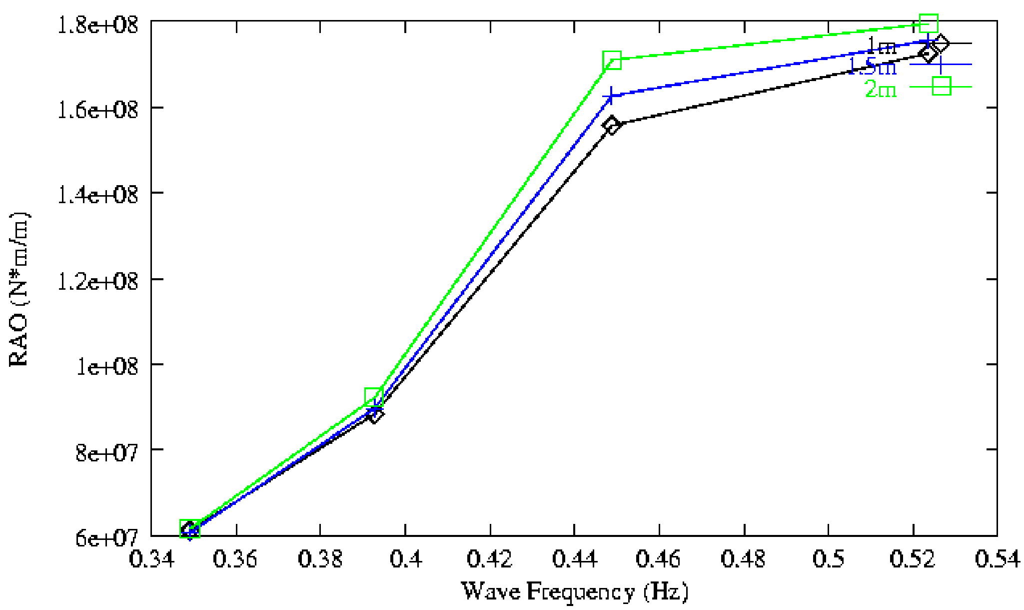

Figure 78 TACS Yaw Moment vs. Wave Freq._225deg (LMSR-TACS)

Appendix 1 Data Organization and Nomenclature

The data files are organized in a hierarchical data structure. There is a directory for each geometric configuration. Inside of each geometry folder, there are directories for the forward speeds. Inside of each forward speed folder, there is a directory for each heading, and inside of each heading folder are the results for all three sea states. These files consist of the LAMP output file (*.out), the point motion file generated by LMPOST (*.map), and the relative motion at a point file, generated by GNU Octave (*.rmap). The formats for the LAMP and LMPOST output files are standards documented in the LAMP documentation. The *.rmap file generated by Octave is a space delimited ASCII text file which contains the relative motion data for each set of points. The first column of this file is the time, and the following 8 columns are x,y,z relative motion, velocity, and acceleration respectively. Each data file is named according to the following convention:

geometry_ssSeaState_Headingdeg_Speedkt.*

Appendix 2 Irregular Figures for 16 Knots

These figures give the irregular wave responses for the 16 knots cases.

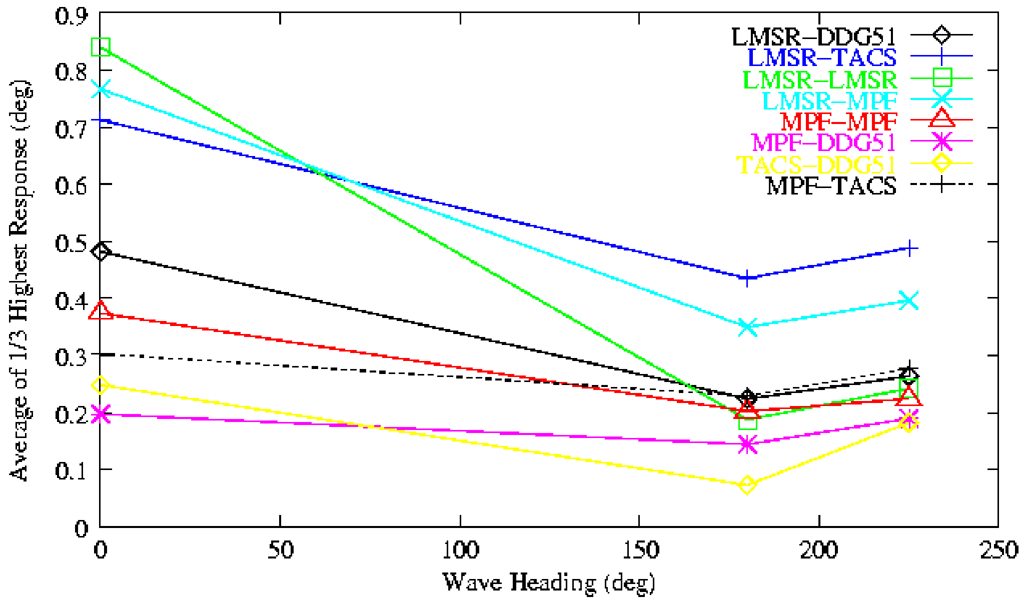

Figure 79 Roll Angle in Sea State 3 at 16 knots For Ship 1

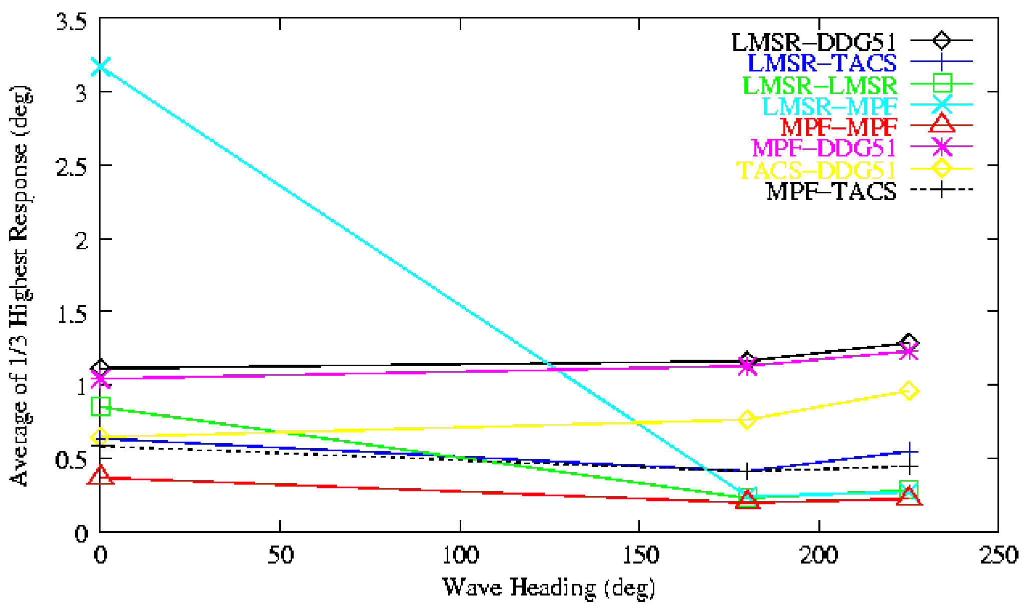

Figure 80 Roll Angle in Sea State 3 at 16 knots For Ship 2

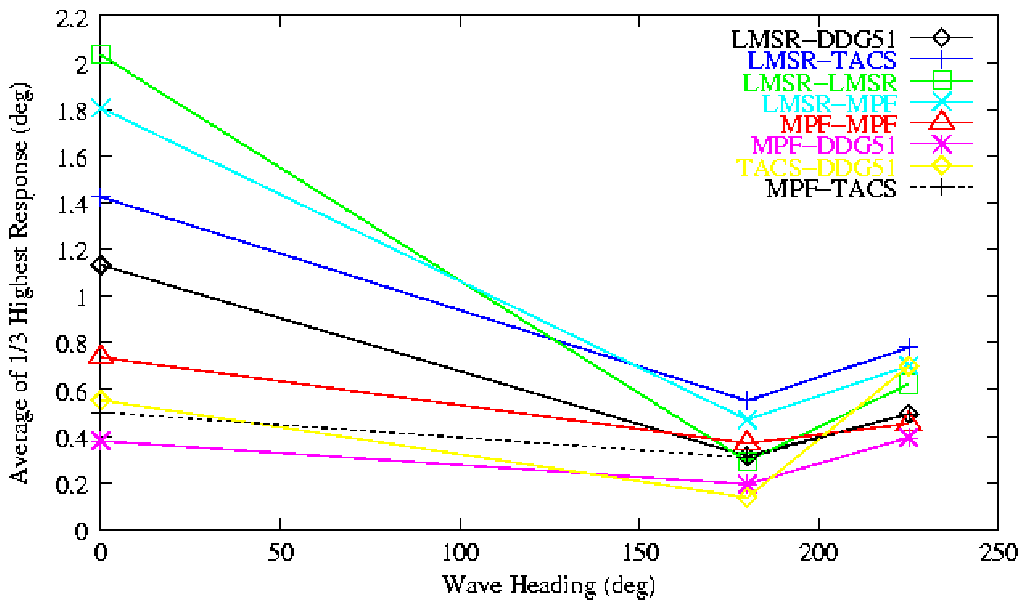

Figure 81 Roll Angle in Sea State 4 at 16 knots For Ship 1

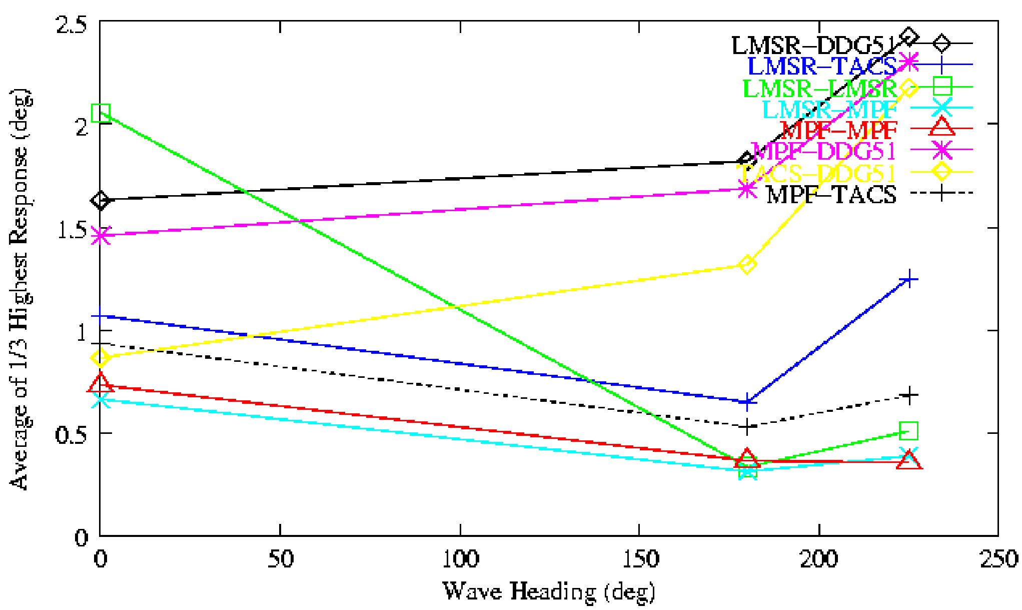

Figure 82 Roll Angle in Sea State 4 at 16 knots For Ship 2

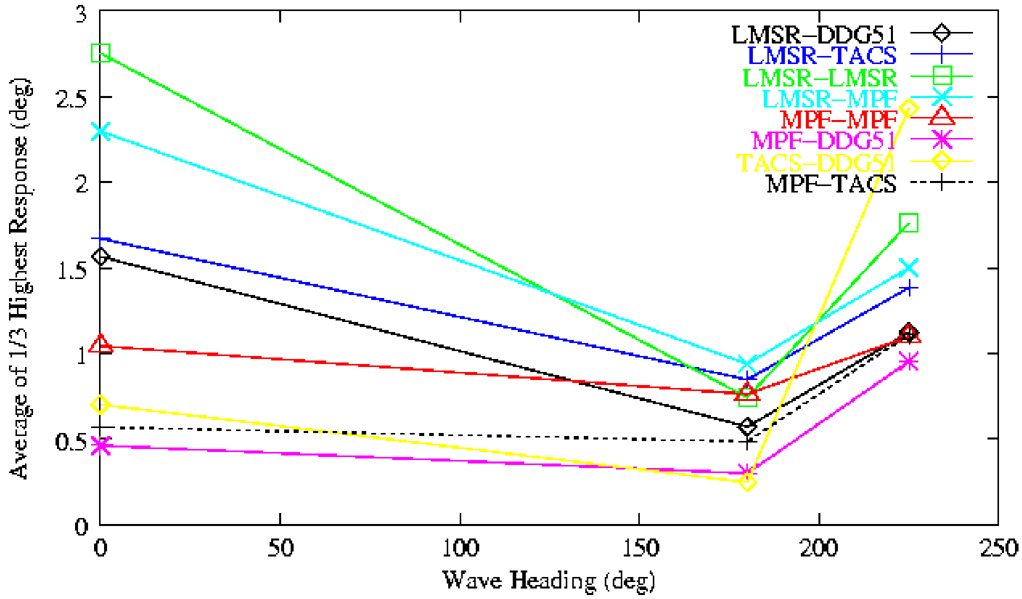

Figure 83 Roll Angle in Sea State 5 at 16 knots For Ship 1

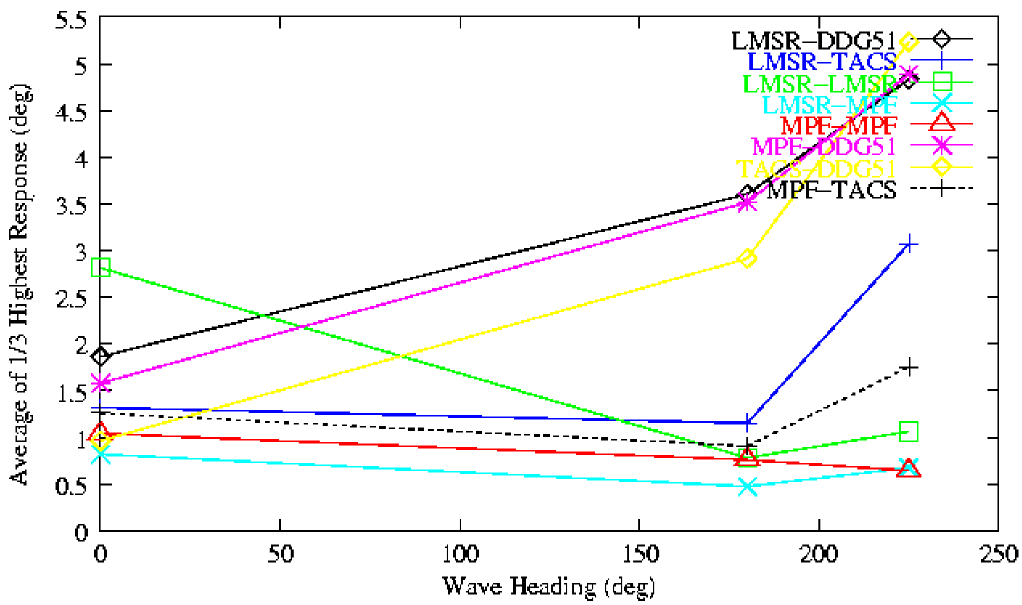

Figure 84 Roll Angle in Sea State 5 at 16 knots For Ship 2

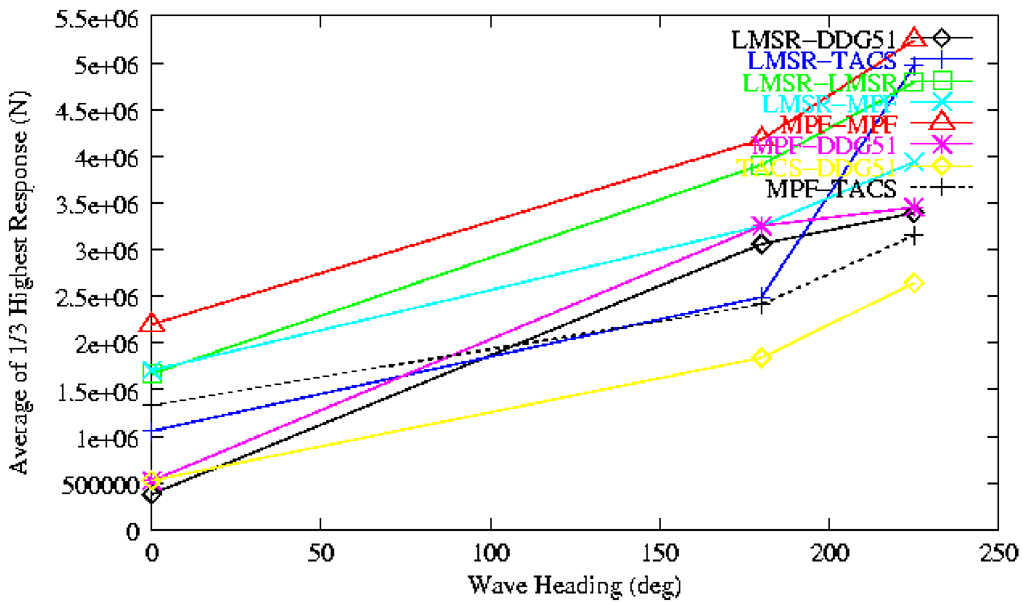

Figure 85 Sway Force in Sea State 3 at 16 knots For Ship 1

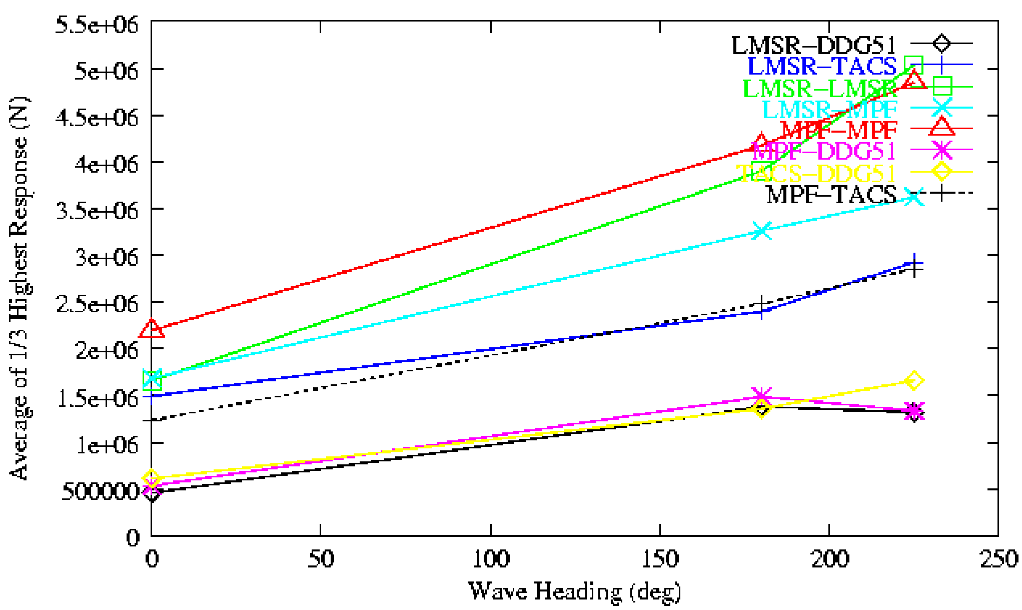

Figure 86 Sway Force in Sea State 3 at 16 knots For Ship 2

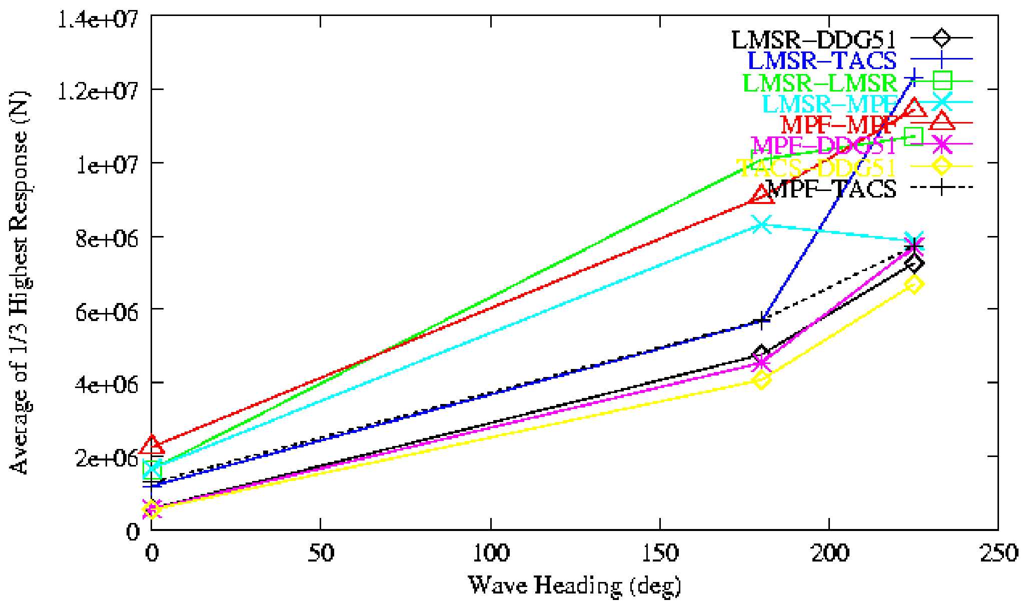

Figure 87 Sway Force in Sea State 4 at 16 knots For Ship 1

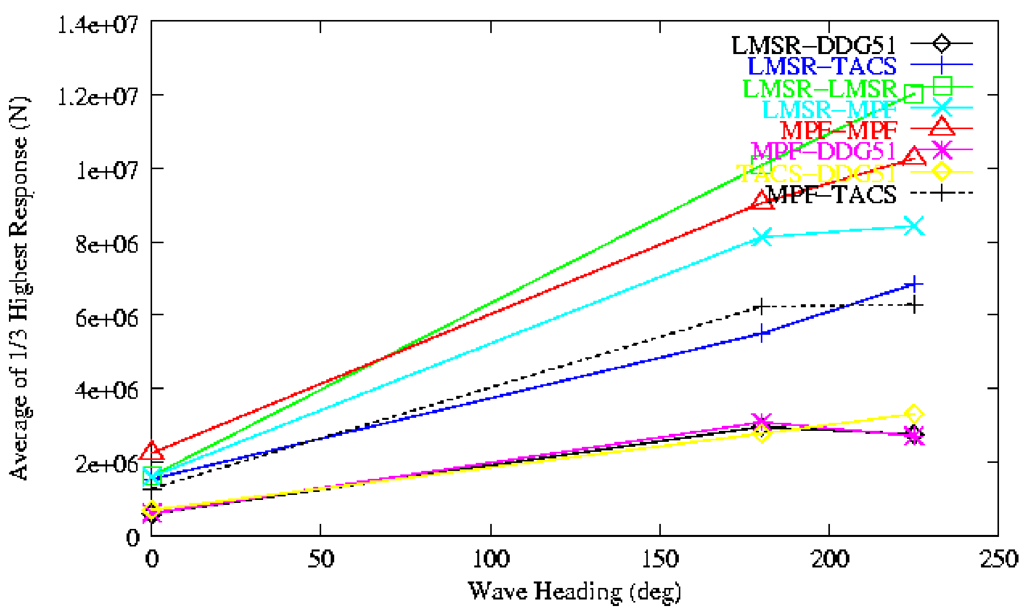

Figure 88 Sway Force in Sea State 4 at 16 knots For Ship 2

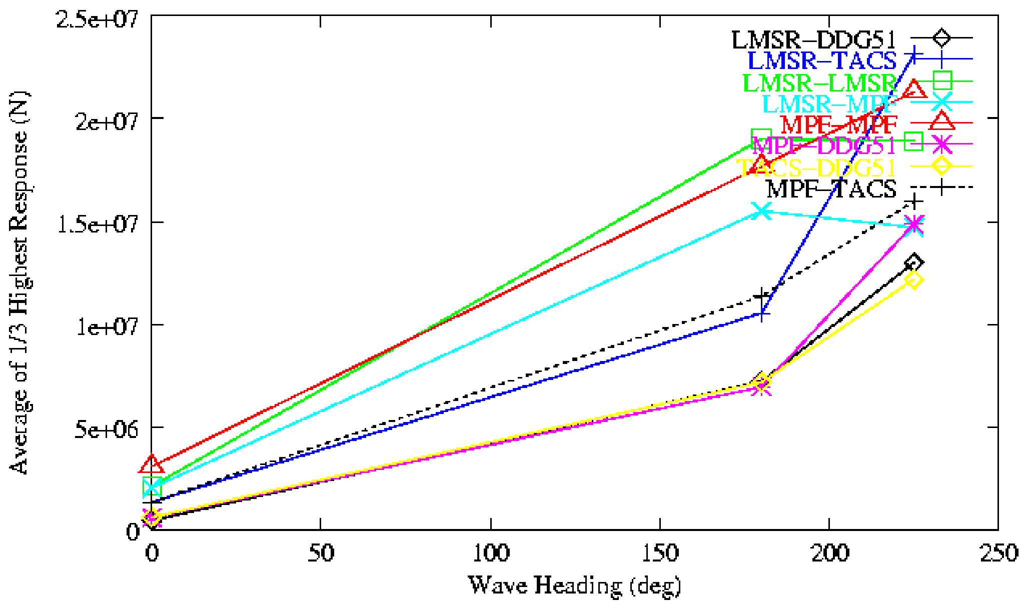

Figure 89 Sway Force in Sea State 5 at 16 knots For Ship 1

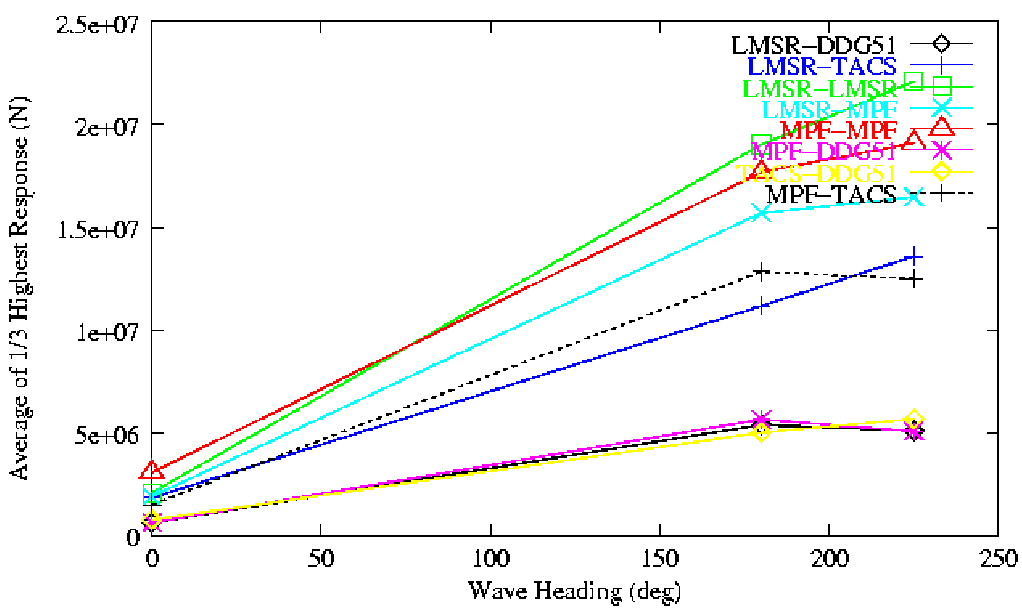

Figure 90 Sway Force in Sea State 5 at 16 knots For Ship 2

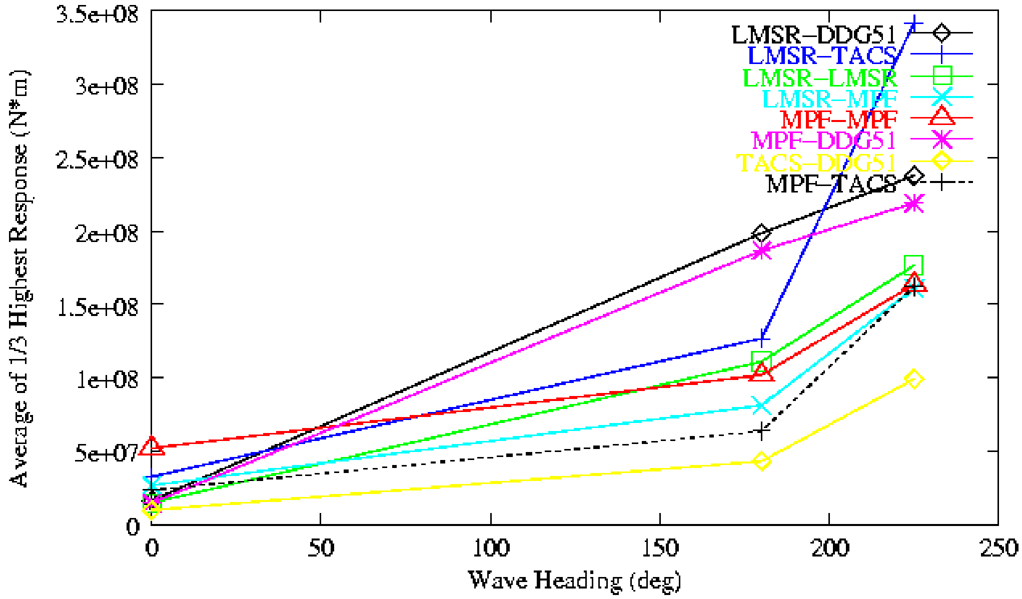

Figure 91 Yaw Moment in Sea State 3 at 16 knots For Ship 1

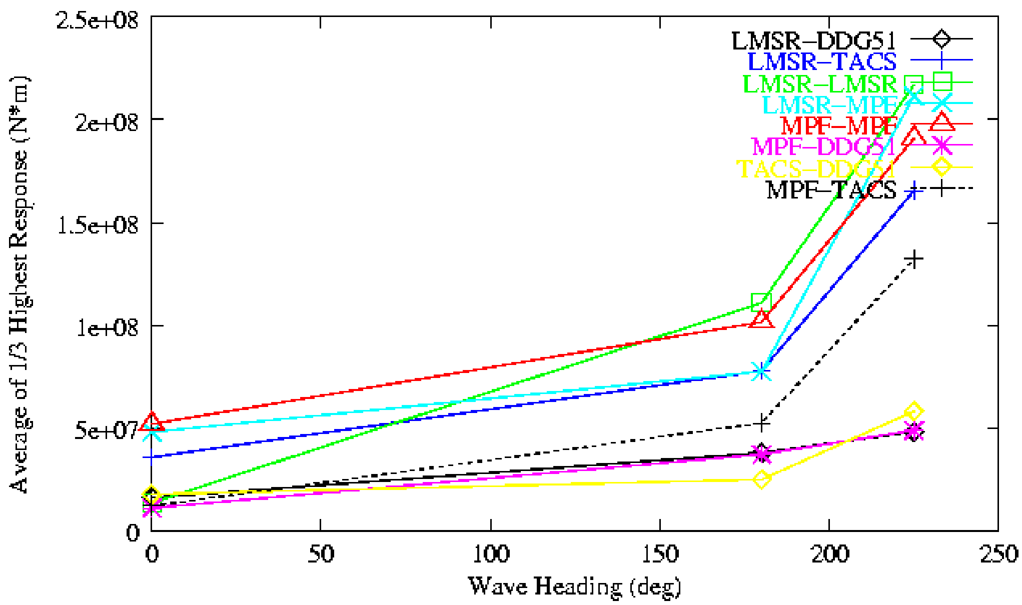

Figure 92 Yaw Moment in Sea State 3 at 16 knots For Ship 2

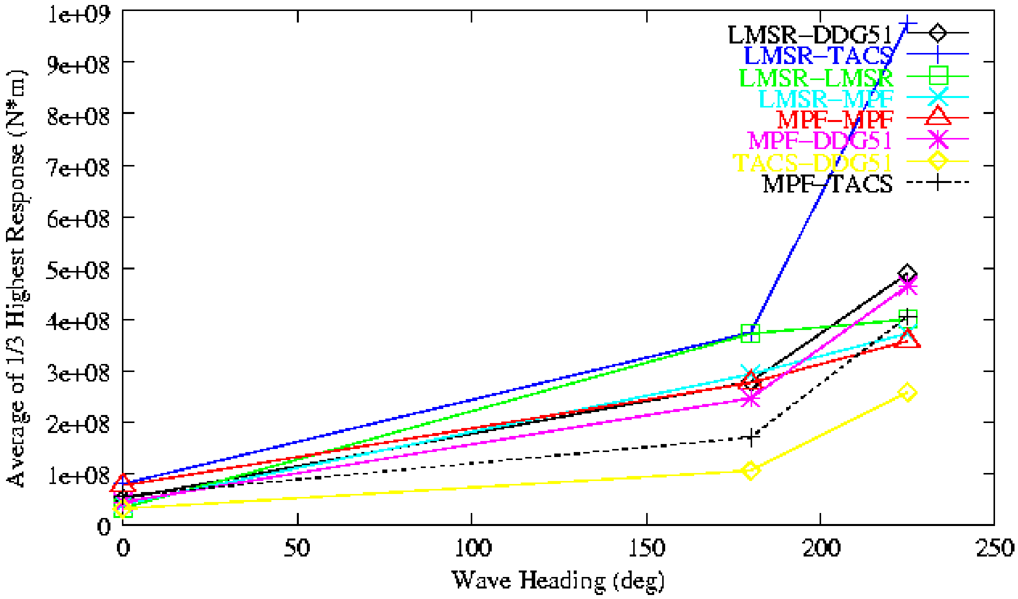

Figure 93 Yaw Moment in Sea State 4 at 16 knots For Ship 1

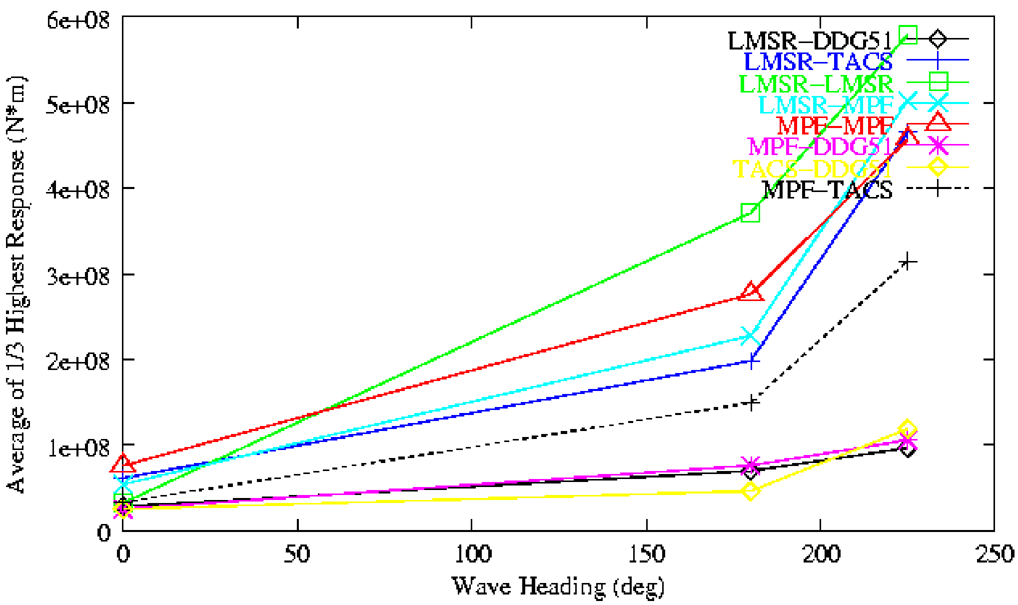

Figure 94 Yaw Moment in Sea State 4 at 16 knots For Ship 2

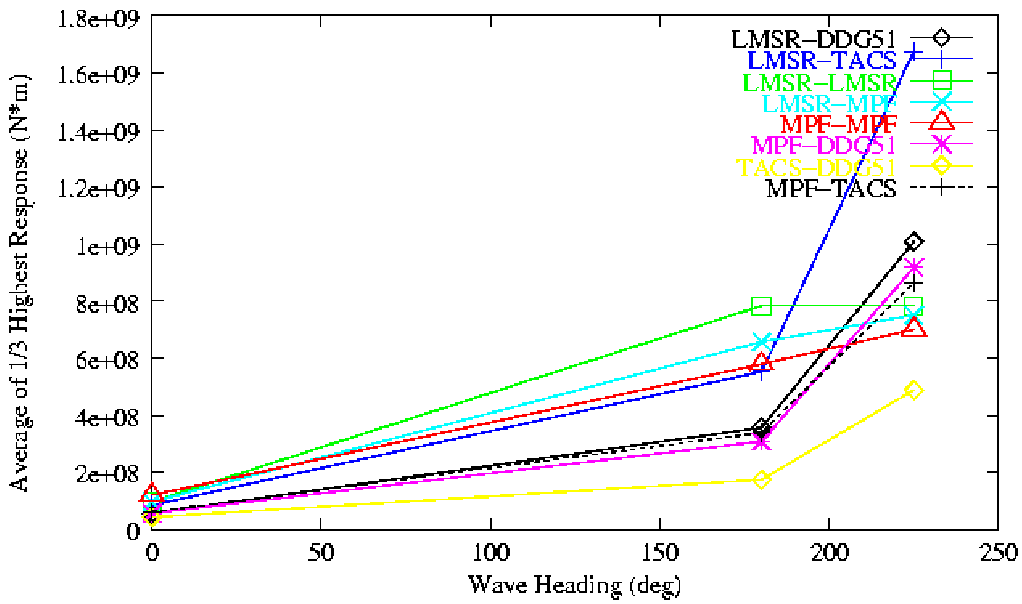

Figure 95 Yaw Moment in Sea State 5 at 16 knots For Ship 1

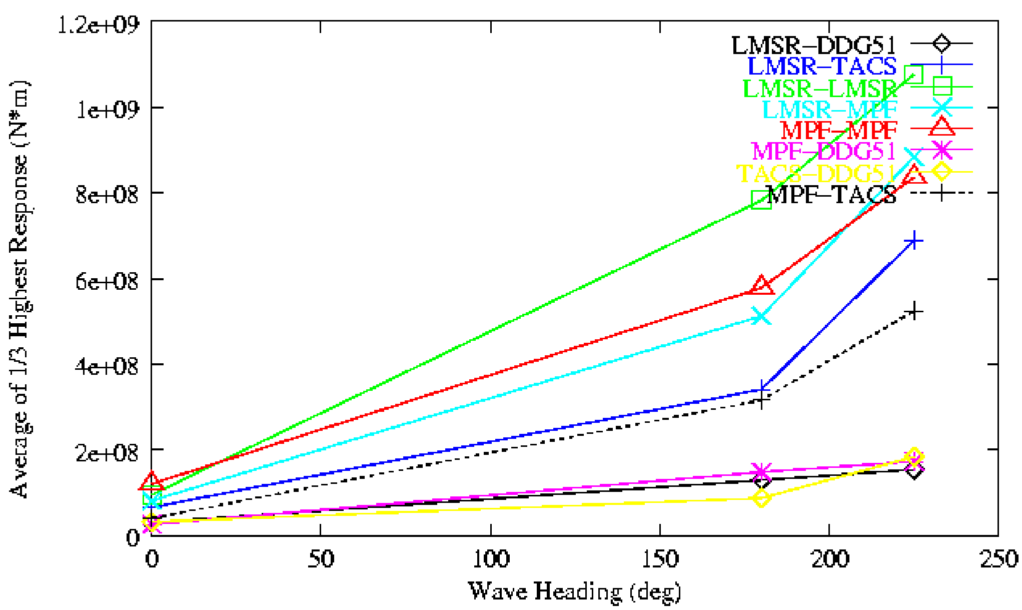

Figure 96 Yaw Moment in Sea State 5 at 16 knots For Ship 2